The Association
of
Engineering and Shipbuilding
Draughtsmen.

GEAR MANUFACTURE.

BOOK 6.

Gear Shaper Cutter Design (Spur Gears).
Gear Shaper Cutter Design (Helical Gears).
Testing of Spur and Internal Spur Gears.
Testing of Spiral and Helical Gears.
Testing of Bevel Gears.

By F. WHITELEY.

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GEAR MANUFACTURE.

By F. Whiteley.

CHAPTER I.

GEAR SHAPER CUTTER DESIGN FOR SPUR GEARS.

As previously stated gear shaper cutters for average external gears have been standardised, a 4·000 pitch circle diameter cutter being used for 6 diametral pitch gears and coarser, and a 3·000 pitch circle diameter cutter for 7 diametral pitch and finer.

For internal gears of course the number of teeth in the cutter depends upon the number of teeth in the gear, and in the majority of instances hub or shank cutters have to be used. The same method is used, however, for the design, and it is also advisable to refer to the machine layout sheets when fixing the bore and overall thickness of the cutter.

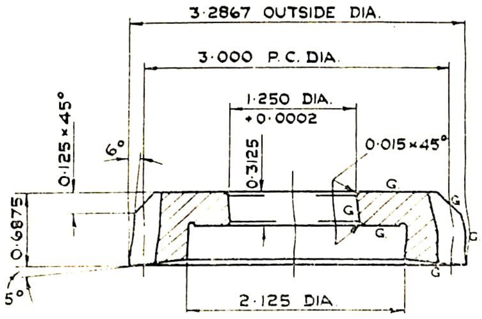
In principle the cutter is similar to a spur gear but with the addition of a side relief of 6° and a front relief of 5°, and for external gears is generally as shown in Fig. 1 in the form of a working drawing.

Assuming that the cutter has to be made suitable for cutting the spur gear, Fig. 1, Book 2, the first point to note is that although the dedendum of the gear to be cut is specified as 0·116, it is necessary to design the cutter to cut a dedendum of 0·125, or 1·250 divided by the diametral pitch.

This is due to the fact that a larger fillet or radius is generated by a cutter of this type than is generated by a hob or rack cutter, consequently more clearance is required in the roots of the gear teeth.

The next step is to determine from the particulars of the gear to be cut, whether a cutter of standard proportions can be used, unless of course this is stated on the gear drawing.

This is done by checking the addendum and the circular tooth thickness of the gear teeth using standard formula, or in the case of corrected gearing it is necessary to obtain the cutter proportions from the particulars of both wheel and pinion as follows :—



CUTTER DATA	
TYPE OF CUTTER	SPUR
PITCH	10 DP
PRESSURE ANGLE	20°
NUMBER OF TEETH	30
CUTTING DEPTH	0·225
CHORDAL ADDENDUM	0·1457
CHORDAL THICKNESS	0·1703
BASE CIRCLE DIAMETER	2·81907

ETCH: CUTTING DEPTH, PITCH
PRESSURE ANGLE, AND
CUTTER NUMBER GS. 901.

MAT^L H.S.S. HARDEN & GRIND
LAP BORE

FIG. 1. WORKING DRAWING OF GEAR SHAPER CUTTER FOR SPUR GEARS.

let A = wheel addendum.
a = pinion addendum.
t_w = circular thickness of wheel.
t_p = circular thickness of pinion.

then cutter is standard if

$$\frac{A + a}{2} = 1/DP \text{ and } t_w + t_p = \pi/DP$$

After having established the addendum, dedendum, and circular tooth thickness of the gear to be cut, and given the pitch and pressure angle the cutter particulars can be calculated as follows :—

when D_c = pitch circle diameter of cutter.
O_c = outside diameter of cutter.
DP = pitch of cutter and gear.
B = addendum of cutter (on plane XX Fig. 2).
A = dedendum of cutter.
c = clearance in roots of cutter teeth.
Z_c = number of teeth in cutter.
ψ = pressure angle of cutter and gear.
y = sharpening allowance.
t_c = circular thickness of cutter tooth.
then $D_c = Z_c/DP$
or $Z_c = D_c \times DP$
and $O_c = D_c + 2\,[B + (y \tan 6°)\,]$

The value for y is given, together with the overall thickness of the cutter in the table below, and it will be noted that the values vary with the pitch of the cutter.

Referring to Fig. 2, it will also be noted that on Section XX the addendum of the cutter is equal to B, and on the outside diameter B has been increased by (y tan 6°) but as the circular tooth thickness is also increased in proportion the cutter will cut the desired size of tooth on either side of Section XX until the base diameter is reached.

Pitch of Cutter	D_c	*Overall thickness of Cutter*	y
6 DP and UP	4·000	0·812	0·275
9 DP to 7 DP	3·000	0·750	0·225
16 DP to 10 DP	3·000	0·687	0·175
Under 16 DP	3·000	0·625	0·125

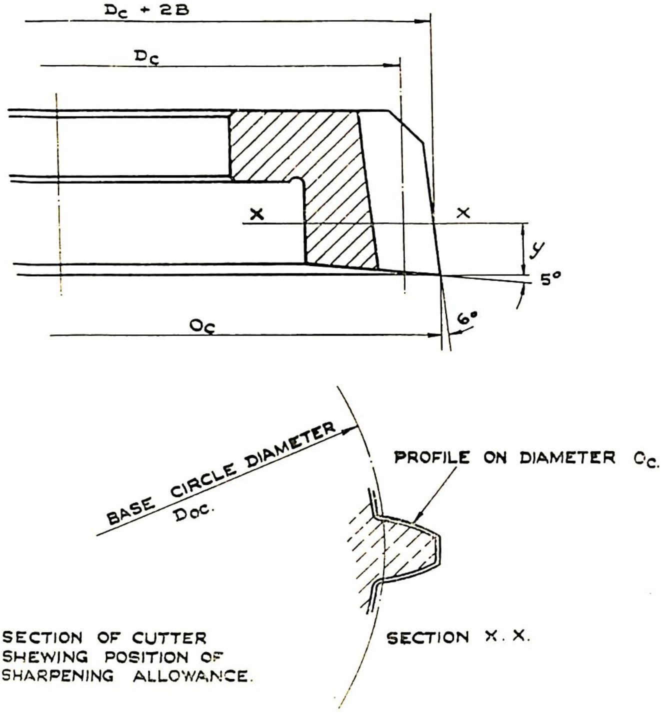


FIG. 2. SECTION OF CUTTER SHEWING POSITION OF SHARPENING ALLOWANCE.

$$\text{then } t_c = \frac{\pi}{2\ \text{DP}} + 2\,(y \tan 6^\circ \tan \psi).$$

and cutter addendum on outside diameter

$$B_o = B + (y \tan 6^\circ).$$

when B_o = cutter addendum on outside diameter of cutter.

The cutter teeth are usually measured using a pair of gear tooth verniers, thus having obtained the values B_o and t_c we proceed as follows :—

Let B_o = addendum of cutter on outside diameter.
t_c = circular thickness of cutter tooth on pitch circle diameter D_c.
t_{cc} = chordal thickness of cutter tooth.
B_{oc} = chordal addendum of cutter tooth.
ϕ = half angle subtended by t_c on diameter D_c.
f = correction factor.

$$\text{then } \frac{t_c}{D_c} = \phi \text{ radians.}$$

when one radian $= 57{\cdot}296°$.

$t_{cc} = \sin \phi \, D_c$

and $B_{oc} = f + B_o$

when $f = D_c/2 \, (1 - \cos \phi)$.

The base diameter D_{oc} is the most important factor governing the design as it controls the shape of the cutter tooth.

No matter what the outside diameter of the cutter becomes through sharpening the cutter by grinding the *front face,* the base diameter remains the same.

then $D_{oc} = D_c \cos \psi$.

when D_{oc} = base circle diameter.

D_c = pitch circle diameter.

ψ = pressure angle.

This base circle diameter is used when checking the accuracy of the tooth profile of the cutter and the maximum error allowed from the base circle to the tip should be not more than 0·0002 inches.

To check the involute tooth shape an involute gear testing machine is used as shown diagrammatically in Fig. 3.

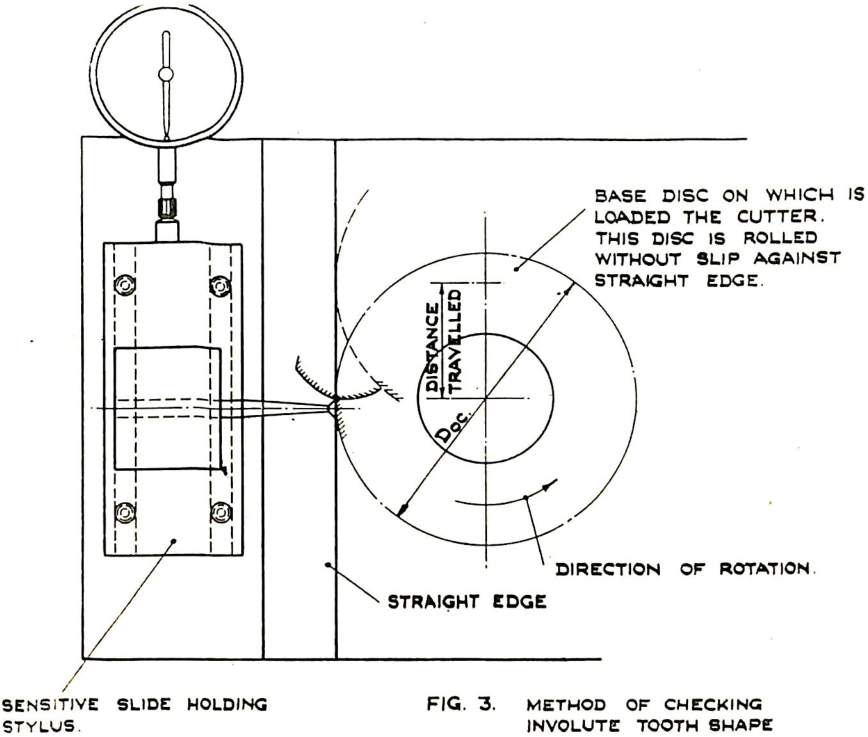

BASE DISC ON WHICH IS LOADED THE CUTTER. THIS DISC IS ROLLED WITHOUT SLIP AGAINST STRAIGHT EDGE.
DISTANCE TRAVELLED
Doc
DIRECTION OF ROTATION.
STRAIGHT EDGE
SENSITIVE SLIDE HOLDING STYLUS.

FIG. 3. METHOD OF CHECKING INVOLUTE TOOTH SHAPE OF GEAR SHAPER CUTTER

A circular disc is used commonly termed a base disc, the diameter of which is made equal to the base circle diameter to a tolerance of minus 0·0002.

Integral with this disc and concentric with it a location diameter to fit the bore of the cutter is arranged, the detail design however depends upon the type of testing machine being used.

Example.

The cutter particulars for a gear shaper cutter to cut the spur gear shown in Fig. 1, Book 2, are calculated below.

Pitch 10 DP (from figure).
Pressure angle 20° (from figure).
$B = 0{\cdot}116$ modified to $0{\cdot}125$ for gear shaper method.
$A = 0{\cdot}100.$
$c = 0{\cdot}030$ to $0{\cdot}040.$

then from table $D_c = 3{\cdot}000.$
$y = 0{\cdot}175.$
and $O_c = 3{\cdot}000 + 2\,[0{\cdot}125 + (0{\cdot}175 \times \tan 6^\circ)\,].$
$= 3{\cdot}2867.$
$t_c = \pi/20 + 2\,(0{\cdot}175 \tan 6^\circ \tan 20^\circ).$
$= 0{\cdot}1704.$
$B_o = 0{\cdot}125 + (0{\cdot}175 \tan 6^\circ).$
$= 0{\cdot}1433.$

overall thickness of cutter from table
$= 0{\cdot}687.$
$t_{cc} = \sin \phi \; 3{\cdot}000.$

when ϕ radians $= \dfrac{0{\cdot}1704}{3{\cdot}000}$
$= 0{\cdot}056822$ radians $= 3^\circ - 15' - 20''.$

then $t_{cc} = \sin 3^\circ - 15' - 20'' \times 3{\cdot}000.$
$= 0{\cdot}1703.$
and $B_{oc} = f + 0{\cdot}1433.$
$= 0{\cdot}0024 + 0{\cdot}1433.$
$= 0{\cdot}1457.$

when $f = \dfrac{3{\cdot}000}{2}\,(1 - \cos 3^\circ - 15' - 20'').$
$= 0{\cdot}0024.$
$D_{oc} = 3{\cdot}000 \times \cos 20^\circ.$
$= 2{\cdot}81907.$

CHAPTER 2.

GEAR SHAPER CUTTER DESIGN FOR HELICAL GEARS.

Gear shaper cutters for helical gears closely resemble the cutters for spur gears described in Chapter 1, with the exception that the cutter teeth are inclined, on a spiral equal to the spiral angle of the gear to be cut.

Another difference is to be found on the front cutting face, as each tooth has to be sharpened normal to the tooth spiral in addition to having a front rake of 5°.

A special fixture is required for this operation for use on a traverse surface grinding machine, whereas a spur cutter is sharpened on a universal grinding machine by simply traversing the wheel along the front face angle of 5°.

As previously stated the pitch circle diameter of the cutters is made as near as possible (depending upon the pitch) to 3·500 inches, while the spiral angle must be the same as the gears, but opposite hand.

Then let Z_c = number of teeth in cutter (nearest whole number).
D_c = pitch circle diameter of cutter.
D_{oc} = base circle diameter of cutter.
ψ_t = transverse pressure angle of cutter and gear.
O_c = outside diameter of cutter.
B = addendum of cutter (on plane XX, Fig. 2).
B_o = addendum of cutter on outside diameter O_c
A = dedendum of cutter.
DP_t = transverse pitch of cutter and gear.
y = sharpening allowance.
t_c = circular thickness of cutter tooth.
t_{cn} = normal circular thickness of cutter tooth.
t_{cc} = normal chordal thickness of cutter tooth.
B_{oc} = normal chordal addendum of cutter tooth.
θ_n = half-angle subtended by t_{cn} on diameter D_{cv}.
D_{cv} = virtual pitch circle diameter of cutter.
f = correction factor.
L_c = lead of cutter.
ϕ_c = spiral angle of cutter.

then $Z_c = 3{\cdot}500 \times DP_t$ to nearest whole number.
or $D_c = Z_c/DP_t$
ϕ_c = same as gear to be cut.

$$L_c = \frac{D_c\,\pi}{\tan\,\phi_c}$$

The sharpening allowance and overall thickness of the cutters vary in accordance with the pitch, and suggested values for these are given in the table below.

Pitch of Cutter	D_C *Approx.*	*Overall Thickness of Cutter*	y
$\frac{6}{8}$ DP and up	3·500	0·875	0·275
$\frac{9}{11}$ DP to $\frac{7}{9}$ DP	3·500	0·812	0·225
$\frac{12}{14}$ DP to $\frac{10}{12}$ DP	3·500	0·750	0·175

Referring to Fig. 2, it will be noted that on the transverse Section XX the addendum of the cutter is equal to B, and on the outside diameter B has been increased by (y tan 6°) but as the circular tooth thickness is also increased in proportion the cutter will cut the desired size of tooth on either side of Section XX until the base diameter is reached.

$$\text{then } O_c = D_c + 2\,[B + (y \tan 6^\circ)\,].$$

$$\text{and } t_c = \frac{\pi}{2\text{ DP}} + 2\,(y \tan 6^\circ \tan \psi_t).$$

$$B_o = B + y \tan 6^\circ.$$

$$D_{oc} = D_c \cos \psi_t.$$

The cutter teeth are measured on the normal plane using a pair of gear tooth verniers, and given t_c.

$$t_{cn} = t_c \cos \phi_c \text{ and}$$

$$D_{cv} = \frac{D_c}{\cos^2 \phi_c}$$

$$\text{then } \theta_n \text{ radians} = \frac{t_{cn}}{D_{cv}} \text{ when one radian} = 57{\cdot}296^\circ.$$

$$t_{cc} = \sin \theta_n \, D_{cv}.$$

$$f = D_{cv}/2\;(1 - \cos \theta_n).$$

$$\text{then } B_{oc} = f + B_o.$$

When checking the accuracy of the tooth profile a base disc corresponding to the base circle diameter D_{oc} is used in the same manner as that described for spur cutters in Chapter 1.

Example.

The cutter particulars for a helical gear shaper cutter to cut a helical gear, right hand spiral, $\frac{6}{8}$ DP, 20° transverse pressure angle, 23° spiral angle are given below, and shown in Fig. 4 in the form of a working drawing.

$$Z_c = 3{\cdot}500 \times 6.$$
$$= 21.$$
$$\phi_c = 23°.$$
$$L_c = \frac{3{\cdot}500 \times 3{\cdot}1416}{0{\cdot}42447} = 25{\cdot}904 \text{ inches left hand.}$$

(*Note.*—This value should be the same lead as the helical guide).

$$O_c = 3{\cdot}500 + 2\ [0{\cdot}1562 + (0{\cdot}275 \tan 6°)\ ].$$
$$= 3{\cdot}8702.$$
$$t_c = \frac{\pi}{2 \times 6} + 2\ (0{\cdot}275 \tan 6° \tan 20°).$$
$$= 0{\cdot}2828.$$
$$B_o = 0{\cdot}1562 + 0{\cdot}275 \tan 6°.$$
$$= 0{\cdot}1851.$$
$$D_{oc} = 3{\cdot}500 \cos 20°.$$
$$= 3{\cdot}2889.$$
$$t_{cn} = 0{\cdot}2828 \cos 23°.$$
$$= 0{\cdot}2603.$$
$$D_{cv} = \frac{3{\cdot}500}{\cos^2 23°} = 4{\cdot}130.$$
$$\theta_n \text{ radians} = \frac{0{\cdot}2603}{4{\cdot}130} = 0{\cdot}063026.$$
$$\text{and } \theta_n = 3° - 36' - 40''.$$
$$t_{cc} = 0{\cdot}06299 \times 4{\cdot}130.$$
$$= 0{\cdot}2601.$$
$$f = \frac{4{\cdot}130}{2}\ (1 - \cos 3° - 36' - 40'').$$
$$= 0{\cdot}0040.$$
$$B_{oc} = 0{\cdot}0040 + 0{\cdot}1851.$$
$$= 0{\cdot}1891.$$

Grinding Cutter Profiles.

To grind the profiles of helical cutters a cutter grinding machine is used which works on the generating principle.

Adjustment is provided on the wheel head to allow it to be set to the spiral angle of the cutter plus or minus the necessary side

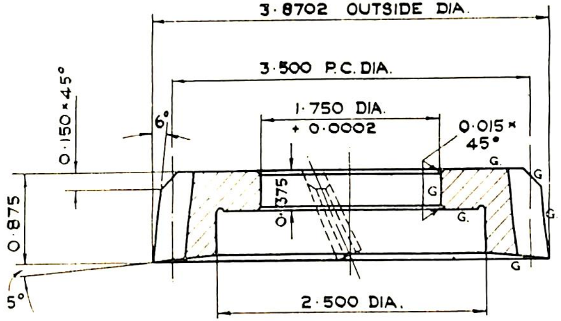


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FIG. 4. WORKING DRAWING OF GEAR SHAPER CUTTER FOR HELICAL GEARS.

CUTTER DATA	
TYPE OF CUTTER	HELICAL
PITCH	6/8 DP
PRESSURE ANGLE	20°
HAND OF CUTTER	LEFT
NUMBER OF TEETH	21
ANGLE OF SPIRAL	23°
LEAD OF SPIRAL	25·904
CUTTING DEPTH	0·281
CHORDAL ADDENDUM	0·1891
CHORDAL THICKNESS	0·2601
BASE CIRCLE DIAMETER	3·2889
PITCH BLOCK DIAMETER	3·528
ETCH :- PITCH, PRESSURE ANGLE, CUTTING DEPTH SPIRAL ANGLE, LEAD AND CUTTER NUMBER HGS 87.	

relief while to obtain the correct involute shape on the transverse plane pitch blocks are provided, the detail design of which varies, depending on the type of machine used.

The wheel head spindle is usually set at a fixed angle of 20° to the horizontal and this angle has to be taken into account when calculating the diameter of the pitch block required.

When D_{oc} = base circle diameter of cutter.
L_c = lead of cutter.
β_o = base lead angle.
ψ_n = angle of wheel head spindle (20°).
β_g = generation angle.
ψ_{tg} = transverse pressure angle of generation.
D_g = pitch diameter of generation.
T = thickness of tapes.
D_p = diameter of pitch block.

and to find the diameter of the pitch block for the helical cutter shown in Fig. 4 when

$$D_{oc} = 3{\cdot}2889 \text{ and } L_c = 25{\cdot}904$$

$$\tan \beta_o = \frac{25{\cdot}904}{\pi \times 3{\cdot}2889} = 2{\cdot}5071 = 68^\circ - 15'$$

$$\psi_n = 20^\circ$$

$$\cos \beta_g = \frac{\cos 68^\circ - 15'}{\cos 20^\circ} = 0{\cdot}39434 = 66^\circ - 47'$$

$$\tan \psi_{tg} = \frac{\tan 20^\circ}{\sin 66^\circ\ 47'} = 0{\cdot}39712 = 21^\circ - 40'$$

$$D_g = 3{\cdot}2889 \sec 21^\circ - 40'$$
$$= 3{\cdot}538$$
$$D_p = 3{\cdot}538 - 0{\cdot}010 \text{ when tape thickness } = 0{\cdot}010$$
$$= 3{\cdot}528$$

A slight adjustment to the grinding wheel angle or pitch block diameter, due to having side relief on the cutters, may be necessary in order to obtain the correct shape on each side of the cutter tooth on the transverse plane.

This shape should, however, be checked on the involute testing machine used for checking spur gear shaper cutters.

CHAPTER 3.

TESTING OF SPUR AND INTERNAL SPUR GEARS.

Gears are tested for the purpose of obtaining readings for :—

(1) Concentricity.
(2) Size of teeth.
(3) Tooth form.
(4) Spacing.
(5) Running.

The five points mentioned are closely related, but all essential to produce high class gearing.

For example, eccentricity can cause varying tooth sizes and inaccuracy of tooth form, thus causing the gears to be noisy under running conditions. Accurate gears however can still be noisy due to bad mounting or being meshed at close centres, and this factor should also have consideration before deciding upon the cause of any trouble should it arise.

Concentricity.

It has previously been mentioned, that before cutting or grinding the teeth in a gear blank the work arbor should be concentric and any end location faces must be square with the arbor.

This also applies to the cutter, as an eccentric cutter or hob can cause varying sizes of teeth, and inaccurate tooth form, therefore this check on the machine before the teeth are cut is of paramount importance.

To obtain the above conditions the equipment supplied for holding the gear blanks should be well designed and durable, made to very close limits of inaccuracy, and wherever possible hardened and ground.

The arbor location should fit the component without being made too tight, which means that the arbor location should be produced within very close limits, and that the arbor should be made no more than 0·0002 to 0·0004 less than the low limit of the hole.

If, therefore, care is taken in designing the equipment it will be found that quite a large amount of trouble often found in gear production has been eliminated.

The same care should be taken in the production of the gear blanks, and if the close limits required cannot be obtained by turning, a grinding operation should be introduced, as the extra time taken will ultimately more than pay for the cost.

For a bored gear blank, before grinding the bore as stated, it is sometimes found necessary to grind the end faces, while in every

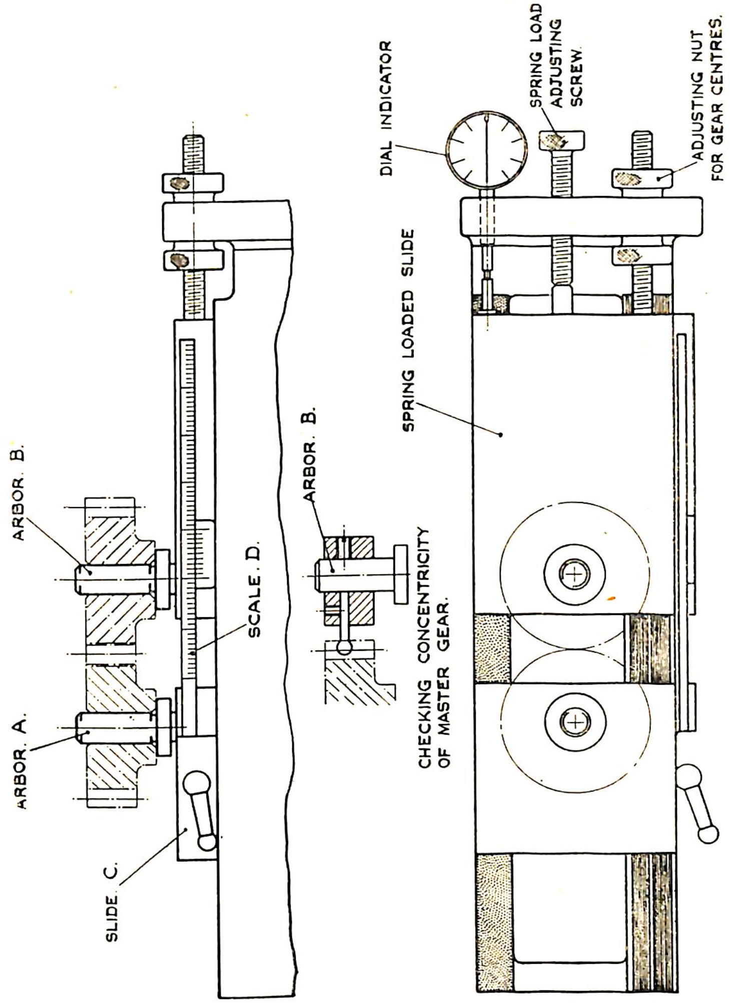

FIG. 5. SKETCH OF TESTING MACHINE FOR SPUR GEARS.

case it is advisable to grind the outside diameter from the bore and thus obtain a concentric blank and ensure that all the teeth are of uniform thickness when measured with gear tooth verniers.

To check concentricity after the gear teeth have been produced, a testing machine is used generally as shown in Fig. 5, this machine also being used to check the gears at running centres.

A master gear is mounted on the arbor A which is located in a spindle mounted in the slide C, this slide being adjustable along the bed of the machine but provision made for locking in any position.

A scale D is secured to the slide in order to obtain the centre distance readings in relation to the spring loaded slide on which is mounted arbor B.

The gear to be tested is mounted on arbor B, and arranged so that the faces of the two gears are in line, the spring loaded slide being adjusted to agree with the running centre distance, this being read off on scale D.

A dial indicator is arranged to contact the spring loaded slide, and with the gears set at the running centre distance this indicator is set at zero.

To check for concentricity a screw is released to bring spring pressure against the slide holding arbor B, thus bringing the two gears tight in mesh, and by rotating the gears, the amount of eccentricity (clock reading) can be observed on the dial indicator.

Autograph recorders are also supplied with these machines, but for general production the dial indicator is generally used.

When observing the movement of the indicator it may be noted that the lever will move slightly, as the gears move from tooth to tooth, and this movement is termed "kick-off," which is caused by interference due to incorrect tooth form.

In the majority of cases, however, it is impossible to eliminate this "kick-off" altogether but an error of more than 0·001 should be investigated.

With regard to the permissible errors of eccentricity, these depend of course on the size of the gear, and also on the allowable backlash, but for all gears up to about 6·000 inches diameter the allowable clock reading should not be more than 0·003.

The master gears used may be taken from those in production, and these can be checked by means of a ball point mounted on arbor B as shown in figure, but it is preferable to manufacture them independant of production, and for durability have them hardened and ground all over.

Master gears should not have any allowance for backlash made on the tooth thickness, or in other words, if two master gears are mounted on the testing machine at running centres they should turn without any eccentricity, kick-off, or backlash.

Size of Teeth.

To measure the size of teeth while the gear is on the gear cutter or grinding machine a pair of gear tooth verniers is used, as previously explained, to check the chordal addendum and chordal thickness dimensions given on the gear drawing.

As a check on these dimensions however, and also to measure fine limit gearing, it is customary to use the roller method of tooth measurement.

For a gear with an even number of teeth two rollers diametrally opposed are inserted in the tooth valleys, and an overall measurement taken with a micrometer or special gap gauge.

For a gear having an odd number of teeth the rollers are disposed as near opposite each other as possible, *i.e.*, one roller is in the tooth valley, adjacent to a tooth, exactly opposite the tooth valley holding the other roller, the overall dimension being given over the two rollers in the same manner as for a gear having an even number of teeth.

There are several methods used for calculating the dimensions over rollers for spur gears. They are as follows :—

(1) Given diameter of rollers to be used or available.
(2) With centre of rollers on pitch circle diameter of gear.
(3) With roller locating on pitch circle diameter of gear.

Case (1) is the one most applicable for both external and internal spur gears, due to the fact that a predetermined size of roller is used, the diameter of which should be about $\frac{1}{16}$ greater than the circular tooth space width on the pitch circle diameter.

By making the rollers slightly greater than the tooth space width it ensures that the rollers will contact on the tooth flanks about the pitch circle, and at the same time allow them to extend beyond the outside diameter of the gear, to facilitate measuring.

To calculate the dimensions over rollers for Case 1, and referring to Fig. 6, we proceed as follows :—

Let D = pitch circle diameter of gear.
p_t = transverse pitch of gear in inches.
t = circular thickness of gear tooth on diameter D.
s = circular width of tooth space on diameter D.
D_o = base circle diameter of gear.
ψ_t = pressure angle of gear.
d = diameter of roller.
θ = angle of roller location.
inv. ψ_t = involute function of angle ψ_t.
inv. θ = involute function of angle θ.
Z = number of teeth in gear.

then $s = p_t - t$.

$$\text{and inv. } \theta = \left(\frac{d}{D_o} + \text{inv. } \psi_t\right) - \frac{s}{D}$$

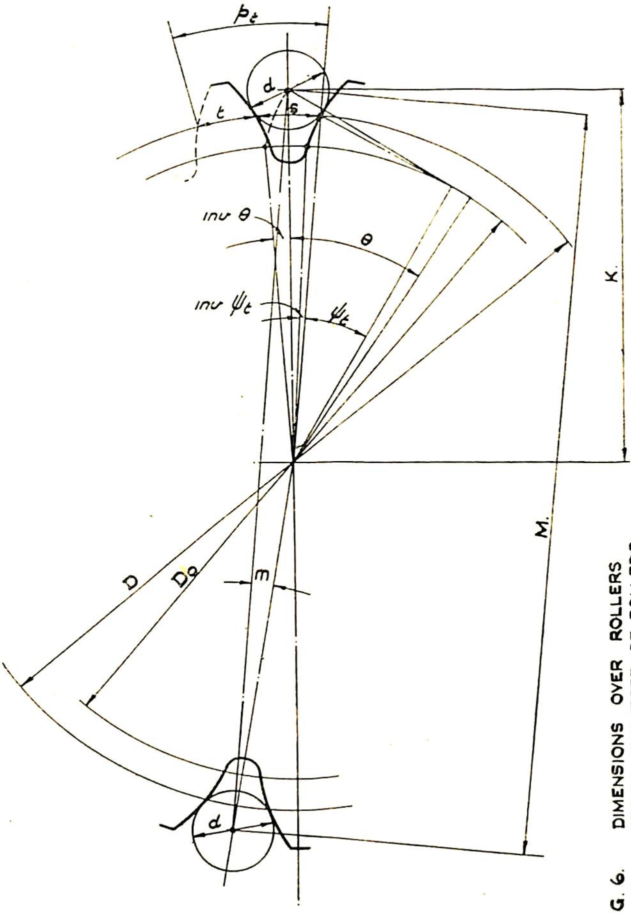

FIG. 6. DIMENSIONS OVER ROLLERS GIVEN DIAMETER OF ROLLERS TO BE USED. (CASE 1)

After obtaining the value of inv. θ in degrees from involute function tables, we have :—

$$2K = \frac{D_o}{\cos\,\theta} \text{ for even number of teeth.}$$

when K = dimension from centre of gear to centre of roller.

For odd number of teeth

$$M = 2K \cos m.$$

when M = dimension to centres of two rollers when number of teeth is odd.

and $m = \dfrac{180}{2Z}$ degrees.

To obtain the allowable tolerance over rollers it is advisable to make two calculations, *i.e.*, with t on high limit and again with t on low limit as shown in the example below, and to obtain the dimension over rollers the roller diameter d must be added to the values 2K or M.

Example (Case 1).

Spur gear 48 teeth, 10 DP, 20° pressure angle, circular thickness of tooth 0·1571 – 0·002/0·004, 4·800 pitch circle diameter, determine dimensions over two rollers 0·1680 diameter.

$s = 0{\cdot}31416 - 0{\cdot}1551$ high limit.
$= 0{\cdot}15906.$
$s = 0{\cdot}31416 - 0{\cdot}1531$ low limit.
$= 0{\cdot}16106.$
$D_o = 4{\cdot}800 \cos 20^\circ.$
$= 4{\cdot}5105.$

$$\text{then inv. } \theta = \left(\frac{0{\cdot}168}{4{\cdot}5105} + \text{inv. } 20^\circ\right) - \frac{0{\cdot}15906}{4{\cdot}800} \text{ high limit.}$$

$$= (0{\cdot}37246 + 0{\cdot}014904) - 0{\cdot}033135.$$

$$= 0{\cdot}019015.$$

$$\theta = 21^\circ - 38' \text{ high limit.}$$

$$\text{inv. } \theta = (0{\cdot}37246 + 0{\cdot}014904) - \frac{0{\cdot}16106}{4{\cdot}800} \text{ low limit.}$$

$$= 0{\cdot}018596.$$

$$\theta = 21^\circ - 28' \text{ low limit.}$$

$$2K = \frac{4{\cdot}5105}{\cos 21^\circ - 38'} \text{ high limit.}$$

$$= 4{\cdot}8522.$$

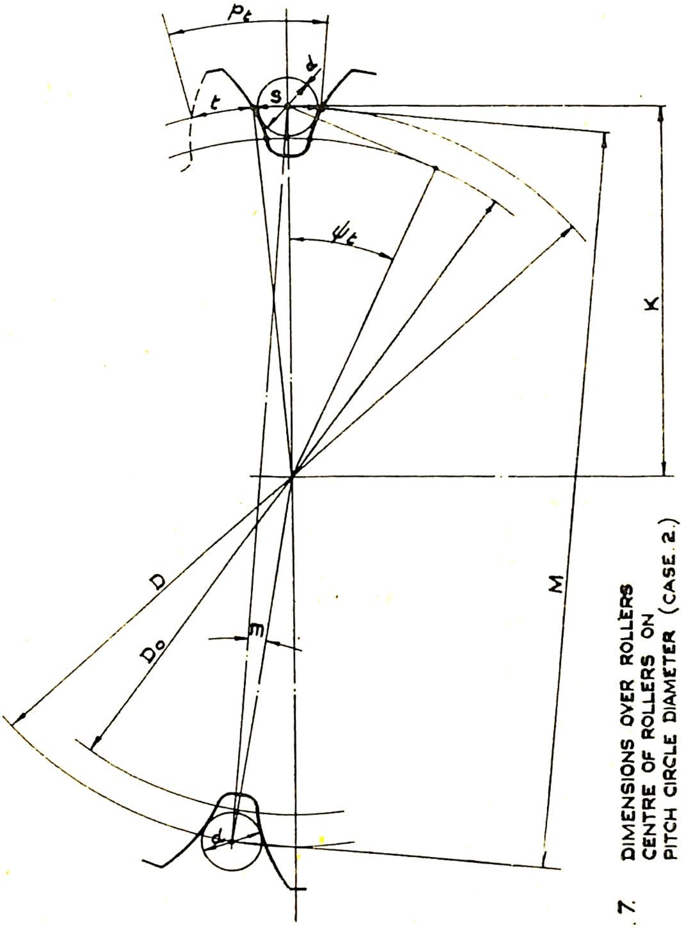

FIG. 7. DIMENSIONS OVER ROLLERS CENTRE OF ROLLERS ON PITCH CIRCLE DIAMETER (CASE 2.)

$$2K = \frac{4{\cdot}5105}{\cos 21^\circ - 28'} \text{ low limit.}$$

$= 4{\cdot}8467.$

and dimension over two rollers 0·168 diameter

$= 5{\cdot}020$ high limit.

$= 5{\cdot}0147$ low limit.

For case (2) the centre of the rollers is positioned on the pitch circle diameter as shown in Fig. 7, and using the same notation as for case (1) we have

$s = p_t - t$ for both high and low limits.

$d = s \cos \psi_t$,, ,, ,, ,, ,,

$D = 2K$ when number of teeth is even.

and $M = 2K \cos m$ when number of teeth is odd.

then dimensions over rollers

$= 2K + d$ when number of teeth is even.

$= M + d$ when number of teeth is odd.

It will be noted that two calculations are required for the diameter of the roller and the dimension over the rollers when a tolerance on the tooth thickness dimension is specified.

Example.

To calculate the dimensions over rollers for Case (2) and using the previous example we have

$s = 0{\cdot}31416 - 0{\cdot}1551$ high limit.

$= 0{\cdot}15906.$

$s = 0{\cdot}31416 - 0{\cdot}1531$ low limit.

$= 0{\cdot}16106.$

$D = 4{\cdot}800.$

then $d = 0{\cdot}15906 \cos 20^\circ$ high limit.

$= 0{\cdot}1494.$

$d = 0{\cdot}16106 \cos 20^\circ$ low limit.

$= 0{\cdot}1513.$

and dimension over rollers.

$= 4{\cdot}800 + 0{\cdot}1494$ high limit over two rollers 0·1494 diameter.

$= 4{\cdot}9494.$

$= 4{\cdot}800 + 0{\cdot}1513.$ low limit over two rollers 0·1513 diameter.

$= 4{\cdot}9513.$

It should be noted that these rollers will not protrude above the outside diameter of the gear.

For Case (3) the condition is that the roller must contact the gear tooth flank on the pitch point as shown in Fig. 8, but it will again be necessary to calculate two diameters of rollers for high and low sizes of tooth.

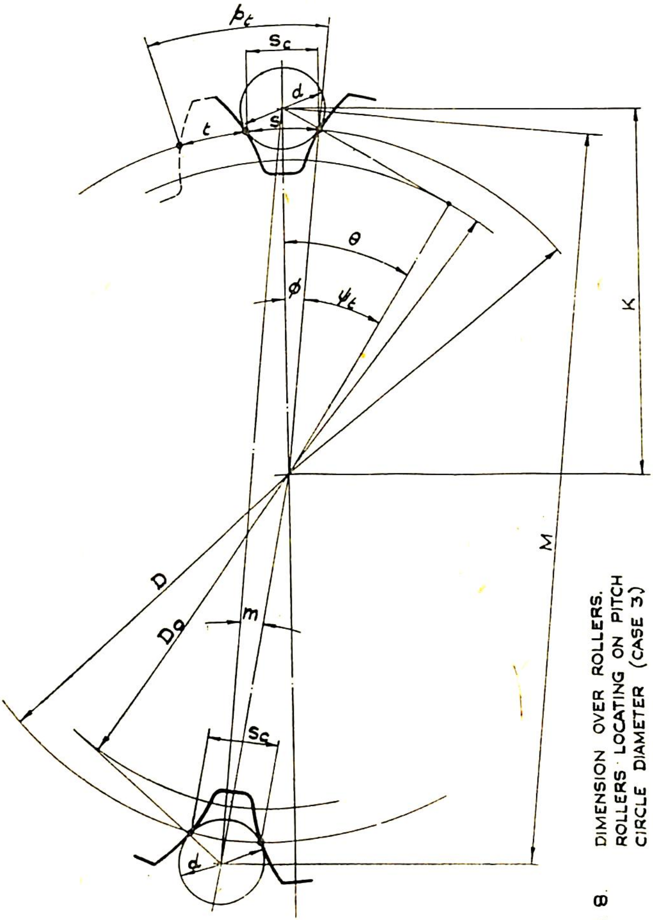

FIG 8. DIMENSION OVER ROLLERS. ROLLERS LOCATING ON PITCH CIRCLE DIAMETER (CASE 3)

To calculate the sizes of rollers required and obtain the dimension over the rollers we proceed as follows, using the same notation, as for cases (1) and (2).

$s = p_t - t$ for both high and low limits.
ϕ radians $= s/D$.
Convert ϕ radians to degrees by multiplying by 57·296 or using tables.
then $s_c = \sin \phi \, D$.
and $\theta = \psi_t + \phi$.

then $d = \dfrac{s_c}{\cos \theta}$ for both high and low limits.

$2K = \dfrac{D_o}{\cos \theta}$ when number of teeth is even.

or $M = 2K \cos m$ when number of teeth is odd.

To values 2K or M must be added the roller diameter d to obtain the dimensions over rollers.

when ϕ = half-angle subtended by s on diameter D.
s_c = chordal width of tooth space.

Example.

Using the previous example the calculations for the dimensions over rollers for case (3) are given below.

$s = 0{\cdot}15906$ high limit of gear.
$s = 0{\cdot}16106$ low limit of gear.

$$\phi \text{ radians} = \frac{0{\cdot}15906}{4{\cdot}800} \text{ high limit.}$$
$$= 0{\cdot}033137 = 1^\circ - 53' - 55''$$

$$\phi \text{ radians} = \frac{0{\cdot}16106}{4{\cdot}800} \text{ low limit}$$
$$= 0{\cdot}033554$$
$$= 1^\circ - 55' - 21''$$

$s_c = \sin 1^\circ - 53' - 55'' \times 4{\cdot}800$ high limit.
$= 0{\cdot}15902.$
$s_c = \sin 1^\circ - 55' - 21'' \times 4{\cdot}800$ low limit.
$= 0{\cdot}16102.$
$\theta = 20^\circ + 1^\circ - 53' - 55''$ high limit.
$= 21^\circ - 53' - 55''.$
$\theta = 20^\circ + 1^\circ - 55' - 21''$ low limit.
$= 21^\circ - 55' - 21''.$

$$d = \frac{0{\cdot}15902}{\cos 21^\circ - 53' - 55''} \text{ high limit.}$$
$$= 0{\cdot}1713.$$

$$d = \frac{0{\cdot}16102}{\cos 21^\circ - 55' - 21''} \quad \text{low limit.}$$

$$= 0{\cdot}1735.$$

$$2K = \frac{4{\cdot}5105}{\cos 21^\circ - 53' - 55''} \quad \text{high limit.}$$

$$= 4{\cdot}8612.$$

$$2K = \frac{4{\cdot}5105}{\cos 21^\circ - 55' - 21''} \quad \text{low limit.}$$

$$= 4{\cdot}8620.$$

and dimension over rollers

$= 4{\cdot}8612 + 0{\cdot}1713$ high limit over two rollers 0·1713 diameter.

$= 5{\cdot}0325.$

$= 4{\cdot}8620 + 0{\cdot}1735$ low limit over two rollers 0·1735 diameter.

$= 5{\cdot}0355.$

It will be found that there are certain disadvantages when using cases (2) and (3), one being that two sizes of rollers are required when a tolerance is specified on the tooth thickness dimension.

Secondly, special size rollers will be required in practically every instance, and for this reason it is recommended that the method described for case (1) should be used for every application, as existing rollers can in the majority of cases be utilised. The rollers used should be hardened, and ground on the outside diameter to a tolerance of 0·0001 if accurate readings are to be obtained, but if difficulty is found in producing the rollers, needle roller bearings make a good substitute.

Internal Spur Gears.

Internal spur gears having involute form of teeth can also be measured using rollers in the same way as for external teeth, excepting of course that the dimension obtained is between the rollers, and not over.

The method used is similar to Case (1) for external spur gears, *i.e.*, when the diameter of rollers is fixed calculate the dimension between rollers.

The diameter of the roller used should be about $\frac{1}{16}$ greater than the circular tooth space width on the pitch circle diameter thus ensuring that the roller will contact the tooth flanks about the pitch circle, and also allow them to protrude beyond the internal diameter of the gear to facilitate measuring.

To calculate the dimensions between rollers, given the roller diameter, and referring to Fig. 9, proceed as follows :—

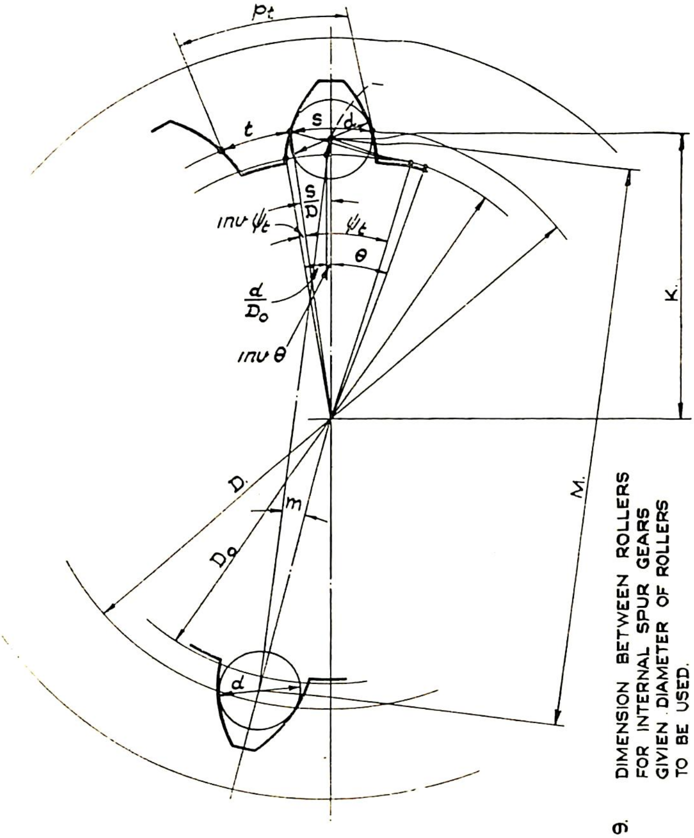

FIG. 9. DIMENSION BETWEEN ROLLERS FOR INTERNAL SPUR GEARS GIVIEN DIAMETER OF ROLLERS TO BE USED.

Let D = pitch circle diameter of gear.
p_t = transverse pitch of gear in inches.
t = circular thickness of gear teeth on diameter D.
s = circular thickness of tooth space on diameter D.
D_o = base circle diameter of gear.
ψ_t = pressure angle of gear.
d = diameter of roller.
θ = angle of roller location.
inv. ψ_t = involute function of angle ψ_t.
inv. θ = involute function of angle θ.
Z = number of teeth in gear.
then $s = p_t - t$.
and inv. $\theta = (\text{inv. } \psi_t + s/D) - d/D_o$.

after obtaining the value of inv. θ in degrees from involute function tables, we have :—

$$2K = \frac{D_o}{\cos\theta} \text{ when number of teeth is even.}$$

when K = dimension from centre of gear to centre of roller,
or when number of teeth is odd
$M = 2K \cos m$
when M = dimension to centres of two rollers when number of teeth is odd.
and $m = 180/2Z$.

To obtain the allowable tolerance between rollers it is advisable to make two calculations, *i.e.*, with t on high limit, and again with t on low limit as shown in the example below, and to obtain the dimensions between rollers the roller diameter d must be deducted from the values 2K or M.

Example.

Internal spur gear, 30 teeth, 16 DP, 20° pressure angle, circular thickness of tooth 0·085 – 0·0015/0·003, 1·875 pitch circle diameter, determine dimensions between two rollers 0·120 diameter.

$s = 0{\cdot}19635 - 0{\cdot}0835$ high limit.
$= 0{\cdot}11285.$
$s = 0{\cdot}19635 - 0{\cdot}0820$ low limit.
$= 0{\cdot}11435.$
$D_o = 1{\cdot}875 \cos 20^\circ.$
$= 1{\cdot}7619.$

$$\text{then inv. } \theta = \left(\text{inv. } 20^\circ + \frac{0{\cdot}11285}{1{\cdot}875}\right) - \frac{0{\cdot}120}{1{\cdot}7619}$$

high limit.

$= (0{\cdot}014904 + 0{\cdot}060186) - 0{\cdot}068108.$
$= 0{\cdot}006982.$
$= 15^\circ - 38'$ high limit.

$$\text{inv. } \theta = \left(\text{inv. } 20^\circ + \frac{0{\cdot}11435}{1{\cdot}875}\right) - \frac{0{\cdot}120}{1{\cdot}7619}$$

low limit.

$$= (0{\cdot}014904 + 0{\cdot}060986) - 0{\cdot}068108.$$
$$= 0{\cdot}007782.$$
$$= 16^\circ - 12' \text{ low limit.}$$

$$2K = \frac{1{\cdot}7619}{\cos 15^\circ - 38'} \text{ high limit.}$$
$$= 1{\cdot}8295.$$

$$2K = \frac{1{\cdot}7619}{\cos 16^\circ - 12'} \text{ low limit.}$$
$$= 1{\cdot}8347.$$

and dimension between two rollers 0·120 diameter

= 1·7095 high limit of tooth thickness.
= 1·7147 low limit of tooth thickness.

Centre Distance Method.

Another method which can be used to measure the size of teeth for both external and internal spur gears, is the centre distance method.

In this case, the testing machine previously described is used, in conjunction with a master gear of known circular tooth thickness, although it is preferable to use a master gear having a circular tooth thickness of nominal size, without any backlash allowance.

The gear to be measured is then mounted on the arbor B, Fig. 5, and the scale set to the centre distance specified on the drawing, when if the gear being tested is correct, the backlash present will be equal to the tolerance given on the gear drawing, or if the tolerance is the same on both wheel and pinion it will be half the total allowable backlash.

This backlash can be converted into terms of change in centre distance, therefore if spring pressure is applied to the slide holding arbor B, the gear being tested will be held close in mesh with the master gear, and the change in centre distance shown on the dial indicator. This value can be calculated, but to avoid making calculations in every case the change in centre distance has been tabulated against the pressure angle, and the backlash in one gear, when meshed with a master gear of nominal tooth thickness as shown in Fig. 10.

If master gears are not available and a gear off the batch is used, then the value of the difference in centre distance according to the pressure angle and backlash in one gear, must be doubled to obtain the correct value.

BACKLASH IN INCHES IN ONE GEAR MESHED WITH MASTER GEAR	$\psi_t = 14\frac{1}{2}^\circ$	$\psi_t = 20^\circ$
	DIFFERENCE IN CENTRE DISTANCE	DIFFERENCE IN CENTRE DISTANCE
0·0010	0·00193	0·00137
0·0015	0·00290	0·00206
0·0020	0·00386	0·00274
0·0025	0·00483	0·00343
0·0030	0·00579	0·00412
0·0035	0·00676	0·00480
0·0040	0·00773	0·00549
0·0045	0·00870	0·00618
0·0050	0·00966	0·00686
0·0060	0·01159	0·00824
0·0070	0·01353	0·00961
0·0080	0·01546	0·01099
0·0090	0·01740	0·01236
0·0100	0·01932	0·01373
0·0110	0·02126	0·01511
0·0120	0·02319	0·01648
0·0130	0·02513	0·01785
0·0140	0·02706	0·01923
0·0150	0·02898	0·02060
0·0160	0·03093	0·02198

FIG. 10. TABLE SHEWING GHANGE IN CENTRE DISTANCE FOR GIVEN BACKLASH.

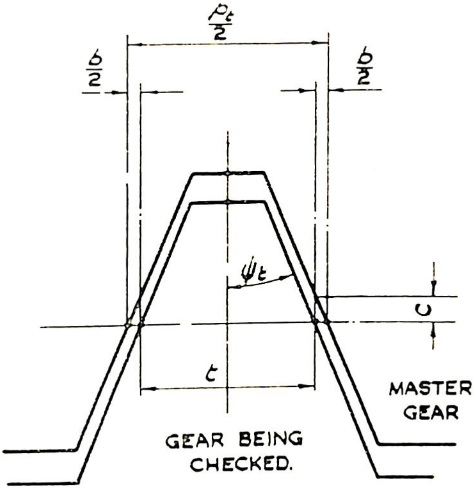

FIG. 11. DIAGRAM SHEWING METHOD OF CALCULATING CHANGE IN CENTRE DISTANCE GIVEN BACKLASH.

To calculate the values given in the table and also to calculate values for gears of other pressure angles we have referring to Fig. 11

ψ_t = pressure angle of gear.
b = backlash in one gear, *i.e.*, when meshed with master gear of nominal tooth thickness.
c = difference in centre distance.
then c = $b/2 \cot \psi_t$.

Example.

Find change in centre distance when backlash in one gear is 0·008 when meshing with master gear and pressure angle is 20°.

$$c = \frac{0{\cdot}008}{2} \cot 20^\circ.$$
$$= 0{\cdot}004 \times 2{\cdot}7475.$$
$$= 0{\cdot}0109.$$

The above method is only approximate, although it is quite accurate enough for practical purposes, but if greater accuracy is required a more detailed calculation will be necessary for the gear pair under consideration.

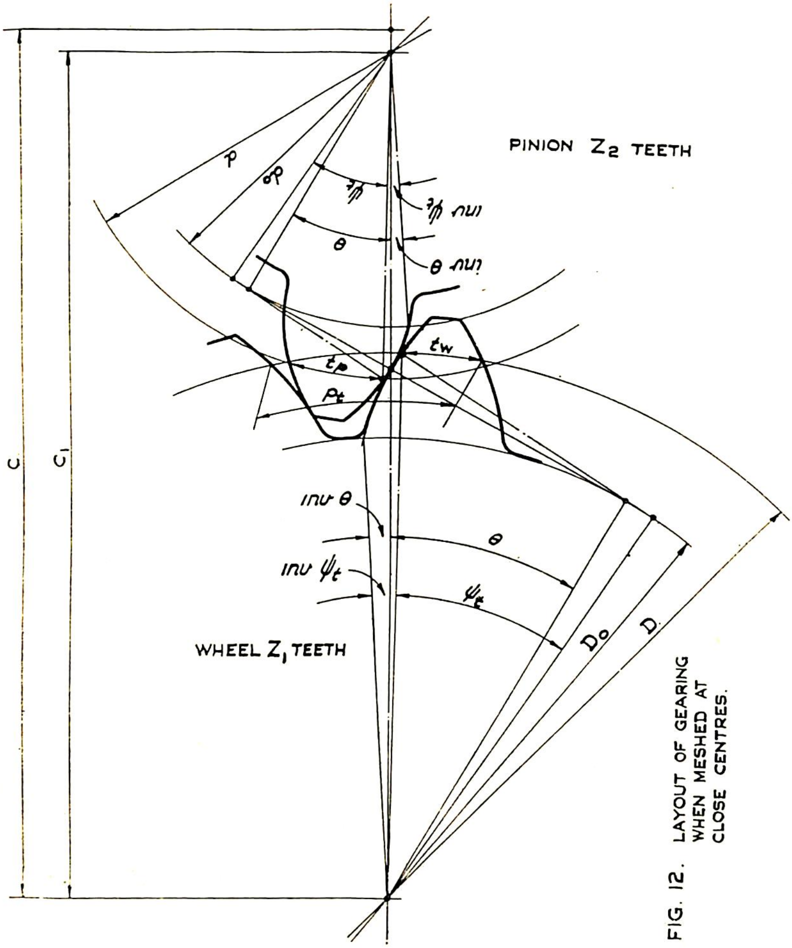

FIG. 12. LAYOUT OF GEARING WHEN MESHED AT CLOSE CENTRES.

To make the calculation it is necessary first to calculate the centre distance at which the gears will mesh tightly, and deduct this value from the original centre distance, then referring to Fig. 12 we have :—

D = pitch circle diameter of wheel.
d = pitch circle diameter of pinion.
Z_1 = number of teeth in wheel.
Z_2 = number of teeth in pinion.
t_w = circular thickness of wheel tooth.
t_p = circular thickness of pinion tooth.
ψ_t = pressure angle at running centres C.
θ = pressure angle at close centres C_1.
C = running centre distance.
C_1 = centres when meshed tightly.
p_t = circular pitch in inches on D or d.

$$\text{then inv. } \theta = \frac{Z_1\ (t_w + t_p) - Z_1\ p_t}{D\ (Z_1 + Z_2)} + \text{inv. } \psi_t.$$

$$\text{when} \quad C = \frac{D + d}{2}$$

$$\text{then} \quad C_1 = \frac{C \cos \psi_t}{\cos \theta}$$

and difference in centres $= C - C_1$.

Example.

To find the change in centre distance when backlash in pinion is 0·008 when meshing with master wheel 2 DP (1·5708 inches) 20° pressure angle and $Z_1 = 40$ $Z_2 = 30$.

then t_w = 0·7854.
t_p = 0·7854 − 0·008 = 0·7774.
D = 20·000.
d = 15·000.

$$\text{then inv. } \theta = \frac{40\ (0{\cdot}7854 + 0{\cdot}7774) - 40 \times 1{\cdot}5708}{20\ (40 + 30)} + \text{inv. } 20^\circ$$

$$= \frac{62{\cdot}5120 - 62{\cdot}8320}{1400} + 0{\cdot}014904.$$

$$= -0{\cdot}0002285 + 0{\cdot}014904.$$

$$= 0{\cdot}014676.$$

$$= 19^\circ - 54'.$$

$$C = \frac{20{\cdot}000 + 15{\cdot}000}{2}$$

$$= 17{\cdot}500.$$

$$\text{then } C_1 = \frac{17{\cdot}500 \cos 20^\circ}{\cos 19^\circ - 54'}$$

$$= 17{\cdot}4888.$$

and difference in centre distance $= 17{\cdot}500 - 17{\cdot}4888.$

$$= 0{\cdot}0112.$$

It will be noted that this value is only 0·0003 greater than the value given in the table, Fig. 10, for the same backlash allowance of 0·008, therefore it will be safe to use the table values and thus avoid making the above calculations in every case.

Distance Across Teeth Method.

Yet another method which can be used to measure the size of external involute spur gear teeth is the distance across teeth method using an ordinary micrometer or vernier.

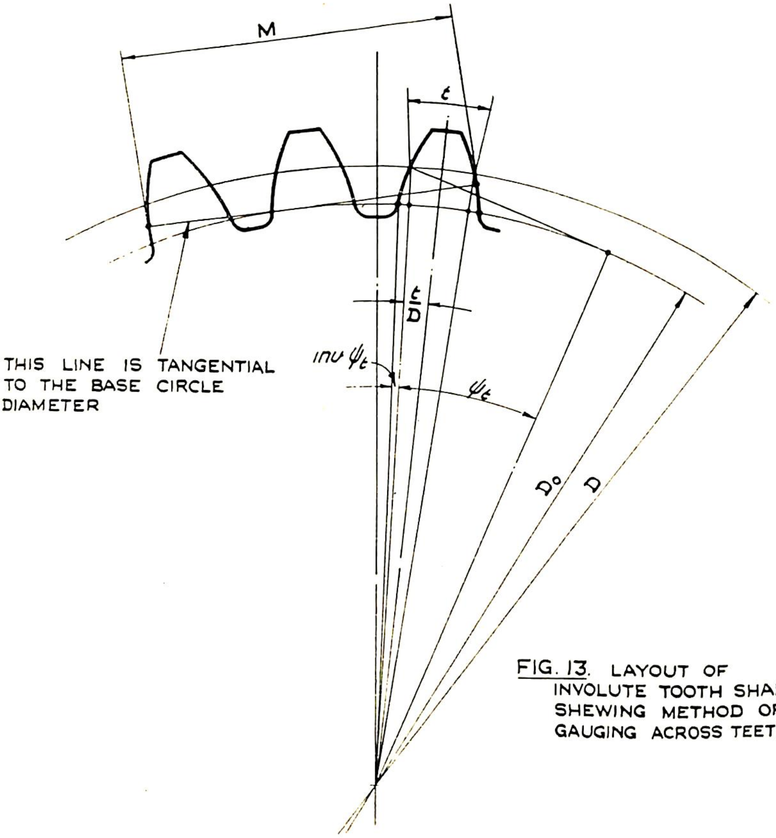

FIG. 13. LAYOUT OF INVOLUTE TOOTH SHAPE SHEWING METHOD OF GAUGING ACROSS TEETH.

This distance is calculated across one or more teeth and can be taken in any position on the tooth flanks as the gauging line is straight, and normal to the involute, therefore it is always tangential to the base circle diameter as shown in Fig. 13.

Referring to the figure

Let M = distance across one or more teeth.
Z = number of teeth in gear.
p_t = transverse pitch in inches on diameter D.
D = pitch circle diameter.
ψ_t = transverse pressure angle on diameter D.
t = circular tooth thickness on diameter D.
D_o = base circle diameter.
N = number of teeth inside gauge.

then $M = [\,(N-1)\,(p_t \cos \psi_t)\,] + D_o\,[\,t/D + \text{inv. } \psi_t\,]$
when the value inv. ψ_t is taken from tables.

and $N = \dfrac{Z\,\psi_t \text{ radians}}{\pi} - (t/p_t) + 1$ to nearest whole number.

Example.

Taking as an example, a spur gear 30 teeth, 10 DP, 20° pressure angle and having a circular tooth thickness of 0·15708, proceed as follows :—

$$N = \frac{30 \times 20^\circ \text{ (radians)}}{\pi} - \left(\frac{0{\cdot}15708}{0{\cdot}31416}\right) + 1$$

nearest whole number

$$= \frac{30 \times 0{\cdot}34906}{\pi} - 0{\cdot}5 + 1.$$
$$= 3{\cdot}3 - 0{\cdot}5 + 1.$$
$$= 3{\cdot}8 \text{ say 4 teeth.}$$

then $M = [\,(4-1)\,(0{\cdot}31416 \cos 20^\circ)\,] + 3{\cdot}000 \cos 20^\circ \left[\dfrac{0{\cdot}15708}{3{\cdot}000} + \text{inv. } 20^\circ\right]$

$$= 0{\cdot}88563 + 2{\cdot}81907 \times 0{\cdot}067264.$$
$$= 0{\cdot}88563 + 0{\cdot}18962.$$
$$= 1{\cdot}0752 \text{ over 4 teeth.}$$

Tooth Size at any Diameter.

The last and final method to be described for measuring involute spur gear tooth shapes is to calculate the circular tooth thickness of the tooth on a series of predetermined diameters.

From the dimensions calculated it is then possible to set out an enlarged drawing of the gear tooth, and check against the actual tooth using an optical projector.

Alternatively the calculations can be used to obtain a series of gear tooth vernier settings by converting the circular dimensions to chordal and checking the gear tooth at the various positions on which the calculations have been made.

With this method, however, it is important that the outside diameter of the gear is exact, as it is used as a datum for setting up on the projector, and also for locating the gear tooth verniers for the chordal addendum reading. To obtain the circular thickness dimensions on another diameter and referring to Fig. 14, we have :—

D = pitch circle diameter of gear.
D_G = diameter on which new tooth thickness is required.
D_o = base circle diameter.
O = outside diameter.
t = circular tooth thickness on diameter D.
t_G = circular tooth thickness on diameter D_G.
t_o = circular tooth thickness on base diameter D_o.
ψ_t = pressure angle on diameter D.
θ = pressure angle on diameter D_G.
inv. ψ_t = involute function of angle ψ_t.
inv. θ = involute function of angle θ.

$$\text{now } \cos\theta = \frac{D\cos\psi_t}{D_G} \text{ or } \frac{D_o}{D_G}$$

$$\text{then } t_G = D_G\,(t/D + \text{inv. }\psi_t - \text{inv. }\theta)$$

$$\text{and } t_o = D_o\,(t/D + \text{inv. }\psi_t)$$

$$\text{or } t = D\,(t_o/D_o - \text{inv. }\psi_t)$$

$$\text{new addendum of gear } A_G = \frac{O - D_G}{2}$$

Should the chordal thickness and addendum be required for the purpose of using gear tooth verniers further calculations will be necessary, but these will be exactly the same as those described in Chapter 1, Book 2, for obtaining pitch line vernier settings, substituting D, t, and A by the values for D_G, t_G and A_G.

Example.

Taking the previous example it is desired to find the tooth thickness on a diameter of 3·100 when D = 3·000.
t = 0·15708.
ψ_t = 20°.
D_o = 2·81907.
O = 3·200.

$$\text{then } \cos\theta = \frac{2{\cdot}81907}{3{\cdot}100} = 0{\cdot}90937 = 24^\circ - 35'.$$

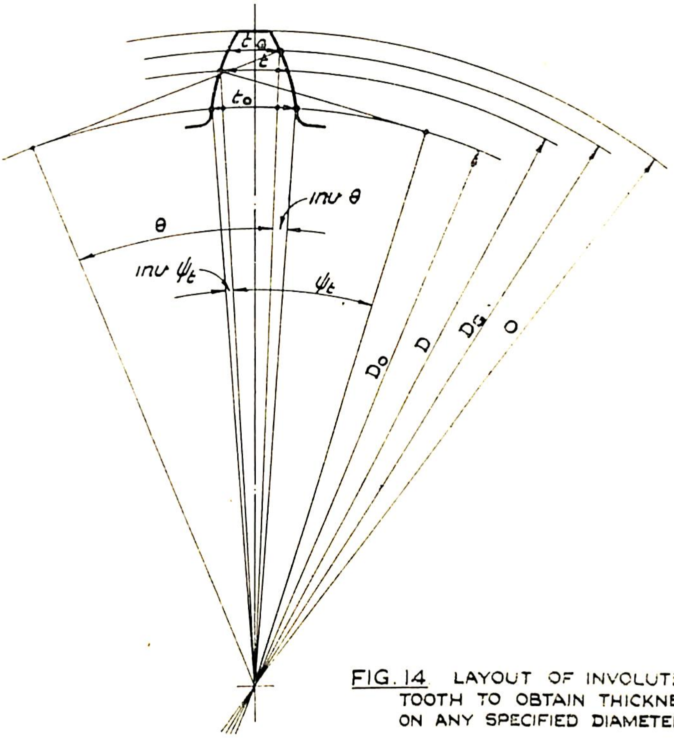

FIG. 14 LAYOUT OF INVOLUTE TOOTH TO OBTAIN THICKNESS ON ANY SPECIFIED DIAMETER.

$$\text{then } t_G = 3{\cdot}100\left(\frac{0{\cdot}15708}{3{\cdot}000} + \text{inv. } 20^\circ - \text{inv. } 24^\circ - 35'.\right)$$

$$= 3{\cdot}100\ (0{\cdot}052360 + 0{\cdot}014904 - 0{\cdot}028424).$$

$$= 3{\cdot}100 \times 0{\cdot}038840.$$

$$= 0{\cdot}120404.$$

$$A_G = \frac{3{\cdot}200 - 3{\cdot}100}{2}$$

$$= 0{\cdot}050.$$

These values should be corrected to chordal dimensions if gear tooth verniers are to be used for checking.

Tooth Form.

The tooth form is very important if two spur gears are to run together smoothly, and for medium pitch gears an involute testing machine is used for checking the flanks of the teeth.

Various types of machines are available but they are all constructed on the same basic principle, *i.e.*, of using a disc the same diameter as the base circle of the gear being tested, and having a stylus arranged to contact the tooth flank, generally as shown in Fig. 3.

Some of these machines are electrically operated, and record the amount of deviation from the true involute, together with the position of the deviations on the tooth flanks.

Only one side of the tooth can be tested at one time. To test the other side the machine is either re-set or the gear turned over.

A point to note with this method of testing is that the tooth flank can only be checked to the base circle. If it extends below this point then the flanks are undercut, and therefore this part of the tooth flank cannot be checked on this machine.

For very small pitch gears, in the teeth of which it is impossible to fit the stylus of the involute testing machine, the centre distance

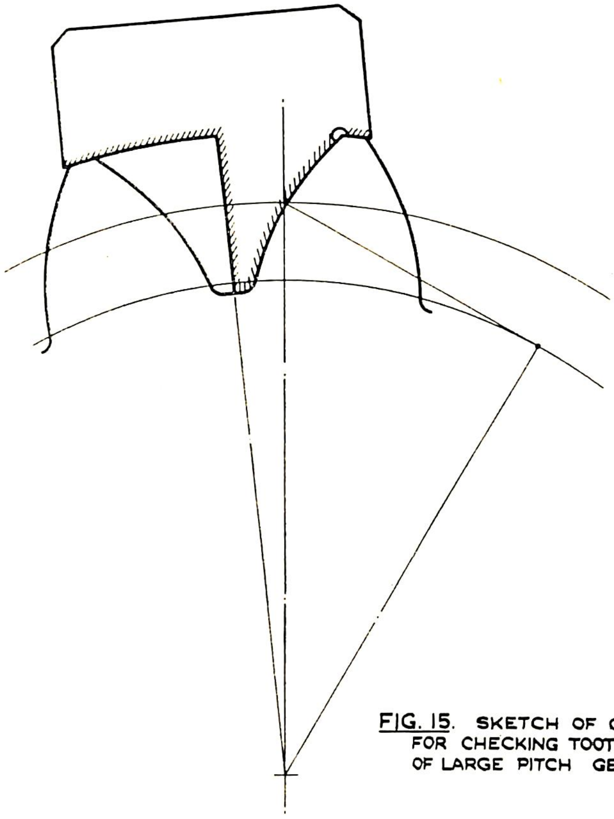

FIG. 15. SKETCH OF GAUGE FOR CHECKING TOOTH SHAPE OF LARGE PITCH GEARS.

machine previously mentioned is used, and any errors in tooth form indicated by the amount of kick-off.

Should it be necessary to check the tooth form of very large gears, which are outside the capacity of the above mentioned machines, it is customary to make a half-tooth space plate gauge from a layout.

This layout is made by the tool draughtsman on a sheet of zinc, and the tool maker produces the gauge to conform to the shape on the layout.

The gauge is arranged to contact the outside diameter of the gear over say two teeth as shown in Fig. 15, and the tooth shape checked to the form on the gauge by holding a light behind the gear tooth and gauge.

Spacing.

Pitch errors should be watched very closely although they are caused by inaccuracies in the dividing wormwheels of the gear cutting machines. It is therefore important that an occasional check should be made on the spacing of the teeth, and for this purpose a mikrotast or similar pitch comparator may be used.

Running.

The final check on gears is of course the running check, either in the assembly or as individual pairs on a testing machine.

If the latter course is adopted a special testing machine is required, having two spindles well mounted, and arranged to receive arbors on which the gears are mounted.

The driven spindle should have facilities for applying a brake in order to load the teeth slightly higher than the load which the gears have to transmit.

The machine should also be installed in a sound proof cabin, thus making it easier to detect any noises should they be present.

CHAPTER 4.

TESTING OF SPIRAL AND HELICAL GEARS.

This type of gear is tested in exactly the same manner as a spur gear previously described, as far as concentricity, spacing, and running are concerned.

With regard to the tooth form, as it is an involute on the transverse plane, or the plane of rotation, the same involute testing machine as used for spur gears is utilised, providing the readings are taken on this plane.

All that is required therefore, is a suitable base disc to locate the gear, the diameter of which is obtained from the formula :—

$D_o = D \cos \psi_t$.

when D_o = base circle diameter.
D = pitch circle diameter.
ψ_t = pressure angle on transverse plane or plane of rotation.

Size of Teeth.

Apart from checking the size of tooth with a pair of gear tooth verniers on the normal plane, it is possible to check both spiral and helical gears by means of rollers, although the formulas given for spur gears will not apply.

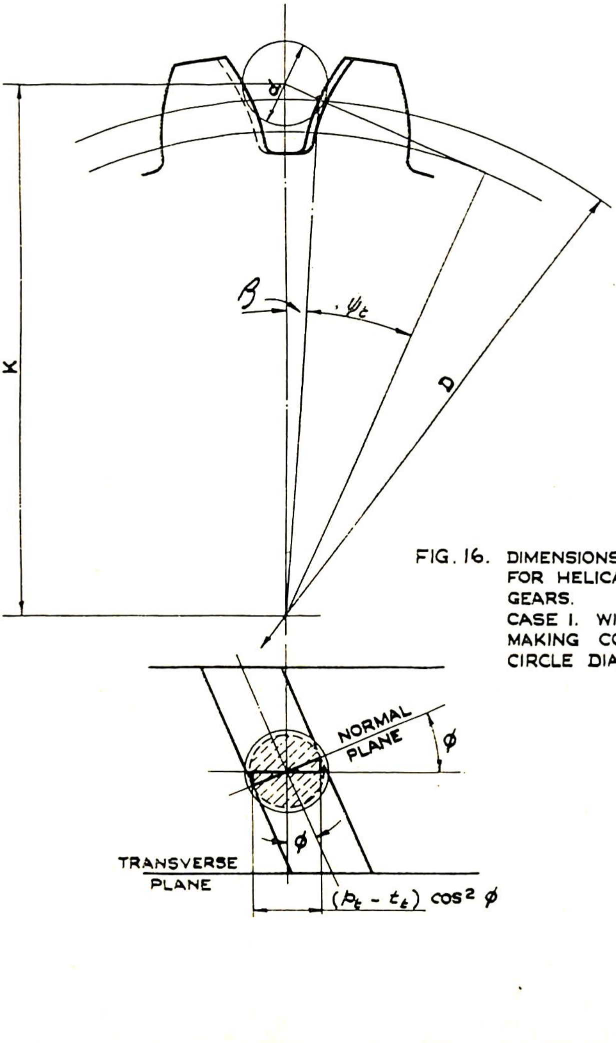

FIG. 16. DIMENSIONS OVER ROLLERS FOR HELICAL AND SPIRAL GEARS.
CASE I. WITH ROLLER MAKING CONTACT ON PITCH CIRCLE DIAMETER.

This is due to the spiral or screw form of the teeth, and is also due to the fact that the calculations are based on the transverse plane, when the rollers actually contact the tooth flanks on the normal plane.

Balls are preferred to rollers but it is very difficult to obtain intermediate sizes, therefore if rollers are used the overall reading should be taken at the minimum distance.

Two methods can be used to calculate dimensions over rollers or balls :—

(1) With rollers making contact on pitch circle diameter, Fig. 16.

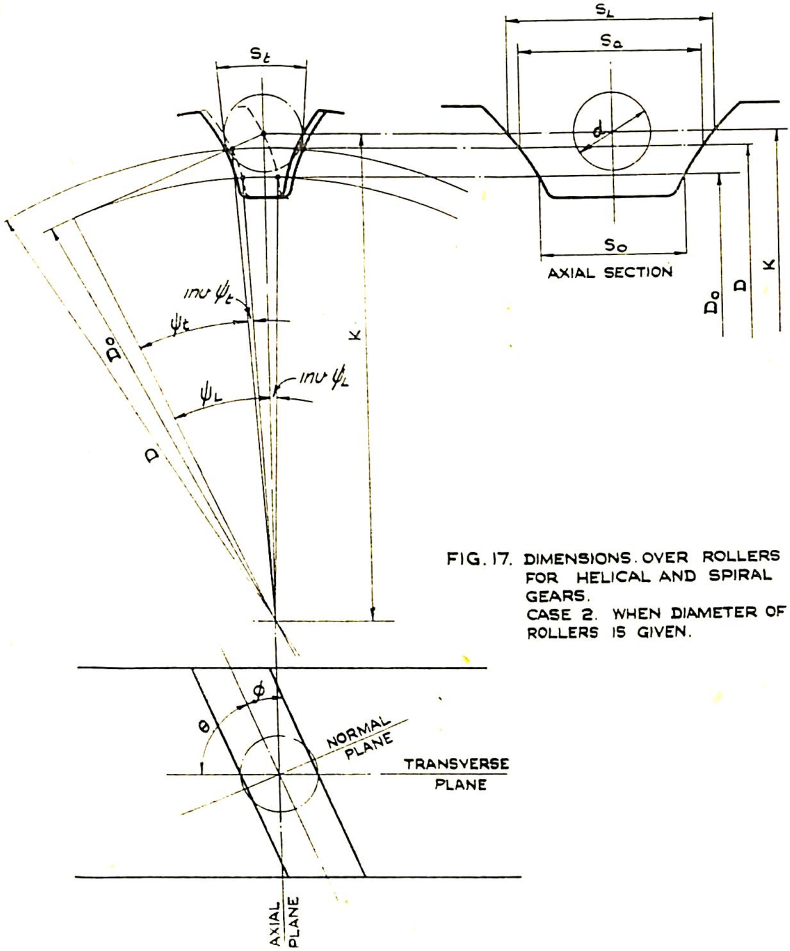

FIG. 17. DIMENSIONS OVER ROLLERS FOR HELICAL AND SPIRAL GEARS.
CASE 2. WHEN DIAMETER OF ROLLERS IS GIVEN.

(2) When diameter of rollers is given, Fig. 17. The second method is the most useful due to the fact that only one size of roller is required, but in any event great care must be taken to ensure that the correct plane, *i.e.*, transverse, normal or axial plane is being used, then referring to figures :—

D = pitch circle diameter.
D_o = base circle diameter.
t_n = circular tooth thickness at D normal plane.
t_t = circular tooth thickness at D transverse plane.
t_a = tooth thickness at D axial plane.
p_n = pitch in inches at D normal plane.
p_t = pitch in inches at D transverse plane.
p_a = pitch in inches at D axial plane.
S_n = circular space width at D normal plane.
S_t = circular space width at D transverse plane.
S_a = axial space width at D axial plane.
S_o = axial space width at D_o axial plane.
S_L = axial space width at K axial plane.
ψ_t = pressure angle at D transverse plane.
ψ_n = pressure angle at D normal plane.
ψ_L = pressure angle at centre of roller.
β = difference between ψ_t and ψ_L transverse plane.
ϕ = spiral angle at D.
θ = $90° - \phi$.
d = diameter of roller.
Z = actual number of teeth in gear.
K = radius to centre of roller from centre of gear.
M = distance between centres of two rollers when number of teeth is odd.
L = lead of spiral on diameter D.
θ_o = base lead angle or lead angle on diameter D_o.

Case 1.

When roller contacts pitch circle diameter D, Fig. 16.

Note.—Two different diameters of rollers will be required, one set for the teeth on top limit and the other for the teeth on low limit, which will necessitate two sets of calculations.

Given D, ψ_t or ψ_n, p_n or p_t, Z, and ϕ.
If ψ_t is given then $\tan \psi_n = \tan \psi_t \cos \phi$.
If ψ_n is given then $\tan \psi_t = \tan \psi_n \sec \phi$.
If p_n is given then $p_t = p_n \sec \phi$.
Or if p_t is given then $p_n = p_t \cos \phi$,

$$\text{and } \beta \text{ radians} = \frac{(p_t - t_t)\cos^2 \phi}{D}$$

$$d = \frac{D \sin \beta}{\cos \phi \cos (\psi_n + \beta)}$$

$$2K = D \cos \beta + d \sin (\psi_n + \beta).$$

Then dimension over two rollers when number of teeth is even

$$= 2K + d$$

and dimension over two rollers when number of teeth is odd

$$= M + d$$

$$\text{when } M = 2K \cos \frac{180^\circ}{2\,Z}$$

Example—Case 1.

Helical gear 20 teeth, $\frac{10}{12}$ DP, 2·000 p.c. dia, 20° transverse pressure angle 23° spiral angle, circular tooth thickness 0·155/0·153.

This gear could be produced by the gear shaper method, therefore the above particulars are based on the transverse plane, although for the purpose of checking the gear with gear tooth verniers a normal chordal thickness dimension would have to be obtained from the dimension 0·155/0·153.

$$\text{then } p_t = 10 \text{ DP} = 0\cdot3146.$$

$$\text{and } \tan \psi_n = \tan 20^\circ \cos 23^\circ = 0\cdot36397 \times 0\cdot92050.$$

$$= 0\cdot33503 = 18^\circ{-}31'{-}20''.$$

$$\cos^2 23^\circ = 0\cdot84732.$$

$$\beta \text{ radians} = \frac{(0\cdot31416 - 0\cdot155) \cos^2 23^\circ}{2\cdot000} \quad \text{high limit}$$

$$= 0\cdot06743 = 3^\circ{-}51'{-}48''.$$

$$\beta \text{ radians} = \frac{(0\cdot31416 - 0\cdot153) \cos^2 23^\circ}{2\cdot000} \quad \text{low limit}$$

$$= 0\cdot06827 = 3^\circ - 54' - 42''.$$

$$d = \frac{2\cdot000 \sin 3^\circ{-}51'{-}48''}{\cos 23^\circ \cos (18^\circ{-}31'{-}20'' + 3^\circ{-}51{-}48'')}$$

$$= \frac{2\cdot000 \times 0\cdot06738}{0\cdot92050 \times 0\cdot92463} \quad \text{high limit.}$$

$$= 0\cdot1583.$$

$$d = \frac{2\cdot000 \sin 3^\circ{-}54'{-}42''}{\cos 23^\circ \cos (18^\circ{-}31'{-}20'' + 3^\circ{-}54'{-}42'')}$$

low limit.

$$= \frac{2\cdot000 \times 0\cdot06822}{0\cdot92050 \times 0\cdot92432}$$

$$= 0\cdot1603.$$

2K (high limit) $= 2{\cdot}000 \cos 3^\circ{-}51'{-}48'' + 0{\cdot}1583 \sin (18^\circ{-}31'{-}20'' + 3^\circ{-}51'{-}48'')$.

$= 2{\cdot}000 \times 0{\cdot}99772 + 0{\cdot}1583 \times 0{\cdot}38084$.

$= 2{\cdot}0557$.

2K (low limit) $= 2{\cdot}000 \cos 3^\circ{-}54'{-}42'' + 0{\cdot}1603 \sin (18^\circ{-}31'{-}20'' + 3^\circ{-}54'{-}42'')$.

$= 2{\cdot}000 \times 0{\cdot}99766 + 0{\cdot}1603 \times 0{\cdot}38160$.

$= 2{\cdot}0564$.

Then dimension over two rollers 0·1583 high limit tooth

$= 2{\cdot}0557 + 0{\cdot}1583$.

$= 2{\cdot}214$.

and dimension over two rollers 0·1603 low limit tooth

$= 2{\cdot}0564 + 0{\cdot}1603$.

$= 2{\cdot}2167$.

Case 2.

When diameter of roller is given, calculate dimensions over rollers.

Given D, ψ_t or ψ_n, p_t or p_n, Z, ϕ and d.

In this case the spiral or helical gear is treated as a worm, having an involute tooth form on the transverse plane or plane of rotation, and referring to Fig. 17.

$\theta = 90^\circ - \phi$.

and $\cos \theta_o = \cos \theta \cos \psi_n$.

$$S_L = \frac{d}{\cos \theta_o}$$

$D_o = D \cos \psi_t$.

should ψ_n be given then $\tan \psi_t = \tan \psi_n \sec \phi$.

$L = \tan \theta \, D \, \pi$.

From this point it will be necessary to make two sets of calculations, one set with tooth thickness on high limit and the other on low limit.

then should S_t be required $S_t = p_t - t_t$.

or should S_n be given $S_n = p_n - t_n$.

or $S_t = S_n \sec \phi$.

then $S_a = S_t \tan \theta$.

and $S_o = S_a - \dfrac{L \text{ inv. } \psi_t}{\pi}$ when value inv. ψ_t is taken from tables.

should ψ_n be given $\tan \psi_t = \tan \psi_n \sec \phi$.

$$S_L = S_o + \frac{L \text{ inv. } \psi_L}{\pi}$$

therefore inv. $\psi_L = (S_L - S_o)\, \pi / L$.

$2K = \dfrac{D_o}{\cos\psi_L}$ when number of teeth is even, then dimension over two rollers when number of teeth is even

$= 2K + d.$

and dimension over two rollers when number of teeth is odd

$= M + d.$

$$\text{when } M = 2K \cos \frac{180^\circ}{2Z}$$

Example—Case 2.

Taking the previous example it is desired to find the dimensions over two rollers for high and low limit tooth sizes, but given the diameter of the rollers of 0·160

when $\psi_t = 20^\circ \qquad \psi_n = 18^\circ{-}31'{-}20''.$

,, $p_t = 0{\cdot}31416 \qquad p_n = 0{\cdot}2892.$

$\theta = 90^\circ - 23^\circ = 67^\circ.$

and $\cos\theta_o = \cos 67^\circ \cos 18^\circ{-}31'{-}20''.$

$= 0{\cdot}39073 \times 0{\cdot}98420.$

$= 0{\cdot}37050 \qquad = 68^\circ{-}15'.$

$$S_L = \frac{0{\cdot}160}{\cos 68^\circ{-}15'} = \frac{0{\cdot}160}{0{\cdot}37050}$$

$= 0{\cdot}4318.$

$D_o = 2{\cdot}000 \cos 20^\circ.$

$= 1{\cdot}8793.$

$L = \tan 67^\circ \times 2{\cdot}000 \times 3{\cdot}1416.$

$= 14{\cdot}802.$

$S_t = 0{\cdot}31416 - 0{\cdot}155$ high limit.

$= 0{\cdot}15916.$

$S_t = 0{\cdot}31416 - 0{\cdot}153$ low limit.

$= 0{\cdot}16116.$

$S_a = 0{\cdot}15916 \tan 67^\circ$ high limit.

$= 0{\cdot}3749.$

$S_a = 0{\cdot}16116 \tan 67^\circ$ low limit.

$= 0{\cdot}3796.$

$$S_o = 0{\cdot}3749 - \frac{14{\cdot}802 \text{ inv. } 20^\circ}{\pi} \qquad \text{high limit.}$$

$= 0{\cdot}3047.$

$$S_o = 0{\cdot}3796 - \frac{14{\cdot}802 \text{ inv. } 20^\circ}{\pi} \qquad \text{low limit.}$$

$= 0{\cdot}3094.$

$$\text{inv. } \psi_L = (0{\cdot}4318 - 0{\cdot}3047) \frac{\pi}{14{\cdot}802} \qquad \text{high limit.}$$

$$= 0{\cdot}1271 \times 0{\cdot}21224.$$
$$= 0{\cdot}026975.$$
$$= 24^\circ - 11'.$$

$$\text{inv. } \psi_L = (0{\cdot}4318 - 0{\cdot}3094)\,\frac{\pi}{14{\cdot}802} \qquad \text{low limit.}$$
$$= 0{\cdot}1224 \times 0{\cdot}21224.$$
$$= 0{\cdot}025978.$$
$$= 23^\circ - 53'.$$

$$2K = \frac{1{\cdot}8793}{\cos 24^\circ - 11'} \qquad \text{high limit.}$$
$$= 2{\cdot}0609.$$

$$2K = \frac{1{\cdot}8793}{\cos 23^\circ - 53'} \qquad \text{low limit.}$$
$$= 2{\cdot}0552.$$

Then dimensions over two rollers 0·160 diameter

$$= 2{\cdot}0609 + 0{\cdot}160 = 2{\cdot}2209 \text{ high limit.}$$
$$= 2{\cdot}0552 + 0{\cdot}160 = 2{\cdot}2152 \text{ low limit.}$$

Centre Distance Method.

The tooth sizes of helical gears can be checked by this method

The centre distance testing machine is used together with a master gear of known circular tooth thickness on the transverse plane.

If the tooth thickness is specified on the normal plane it should be converted in terms of the transverse plane, when

t_t = $t_n \sec \phi$.
t_t = circular tooth thickness transverse plane.
t_n = circular tooth thickness normal plane.
ϕ = spiral angle.

The same remarks also apply to the pressure angle should ψ_n be specified, then

tan ψ_t = tan ψ_n sec ϕ.
when ψ_n = basic normal pressure angle.
ψ_t = pressure angle transverse plane.
ϕ = spiral angle.

Having obtained the values for ψ_t and t_t, the centre distance at which the gears will mesh tightly can be calculated by using the same formula as used for spur gears described in Chapter 3, and deducting this value from the original centre distance the change in centre distance is obtained.

Checking Lead.

The lead of spiral should also be checked for high class gearing, and various types of lead checking machines are available.

The basic principle employed is that for one complete revolution of the gear the axial movement of the stylus engaging the tooth flank must equal the lead in inches.

Therefore for a given rotary movement the axial movement can be calculated thus :—

let p = circumferential movement of gear.
F = axial movement of stylus.
L = lead of spiral.
ϕ = spiral angle.
D = pitch circle diameter.

$$\text{then } \frac{\pi D}{p} = \frac{L}{F} \text{ or } p = \frac{\pi DF}{L}$$

The two motions are usually combined by means of a sine bar arrangement, the sine bar being set at an angle equal to the spiral angle, while the stylus is set at a distance equal to $D/2$ from the centre of the gear.

Distance Over Teeth.

This method can be used for spiral and helical gears but its scope is somewhat limited due to the fact that the face width of the gear must be of sufficient width to allow the measuring faces of the micrometer or vernier to contact the tooth flanks.

Before this fact can be established, however, it is first of all necessary to obtain the distance over teeth dimension and this is calculated as follows :—

when M = distance across one or more teeth.
Z = number of teeth in gear.
p_t = transverse pitch in inches on diameter D.
D = pitch diameter.
ψ_t = transverse pressure angle on diameter D.
t = circular tooth thickness at D transverse plane.
D_o = base circle diameter.
N = number of teeth inside gauge.
ψ_n = normal pressure angle on diameter D.
ϕ = spiral angle on diameter D.
p_n = normal pitch in inches.
ϕ_o = spiral angle on base diameter D_o.
t_n = circular tooth thickness at D normal plane.

Given ψ_n then $\tan \psi_t = \tan \psi_n \sec \phi$.
Given p_n then $p_t = p_n \sec \phi$.
and $\tan \phi_o = \cos \psi_t \tan \phi$.

$$\text{then } N = \frac{\psi_t}{180} \times \frac{Z}{\cos\phi_o \cos\phi} + 1$$

$$\text{and } M = p_n \cos\psi_n \, [t_n/p_n + (N-1) + (Z/\pi) \text{ inv. } \psi_t]$$

The above measurement can be used providing that the face width of the gear is not less than the value.

$$M \sin\phi_o.$$

Example.

Helical gear 30 teeth, $\psi_n = 20°$ $\phi = 30°$ $p_n = 0{\cdot}31416$ (10 DP)

$t_n = 0{\cdot}157.$

$$\tan\psi_t = \tan 20° \sec 30°.$$
$$= 22° - 48'.$$
$$\tan\phi_o = \cos 22° - 48' \tan 30°.$$
$$= 0{\cdot}53223 = 28° - 01'.$$

$$\text{then } N = \frac{22°-48'}{180} \times \frac{30}{\cos 28°-01 \cos 30°} + 1$$
$$= 0{\cdot}126 \times 39{\cdot}2 + 1.$$
$$= 5{\cdot}939 \text{ say } 5 \text{ teeth.}$$
$$M = 0{\cdot}31416 \cos 20°$$
$$\left[\frac{0{\cdot}157}{0{\cdot}31416} + (5-1) + \frac{30}{\pi} \text{ inv. } 22°-48'\right]$$
$$= 0{\cdot}2952\,[4{\cdot}500 + 0{\cdot}2141].$$
$$= 1{\cdot}3916.$$

In order to use this dimension the face width must not be less than $1{\cdot}3916 \sin 28° - 01'.$

$$= 0{\cdot}6533.$$

Double Helical Gears.

These gears are checked in exactly the same manner as helical gears as the tooth profiles are the same on each helix.

Care should be taken, however, before making any calculations in deciding where the pitch, pressure angle and tooth thickness is specified, as if they are given on the normal plane, corresponding values should be obtained on the transverse plane.

CHAPTER 5.

TESTING OF BEVEL GEARS.

Bevel gears, both straight tooth and spiral, are the most difficult gears to check, due to the fact that the pitch cones of two mating gears should meet at a common apex point, which in practice is non-existent.

To obtain this condition as near as practicable the outside diameter, and tip distance, are machined to very close tolerances to ensure that the cone distance is also produced as accurately as possible. This then is the very first essential, for if the above mentioned dimensions depart from those specified, false readings will be obtained when checking the tooth sizes.

To check the blank it is advisable first to manufacture a very accurate dummy blank in the tool room, its dimensions conforming in every way to the high limit dimensions of the gear blank, just prior to the gear cutting operation.

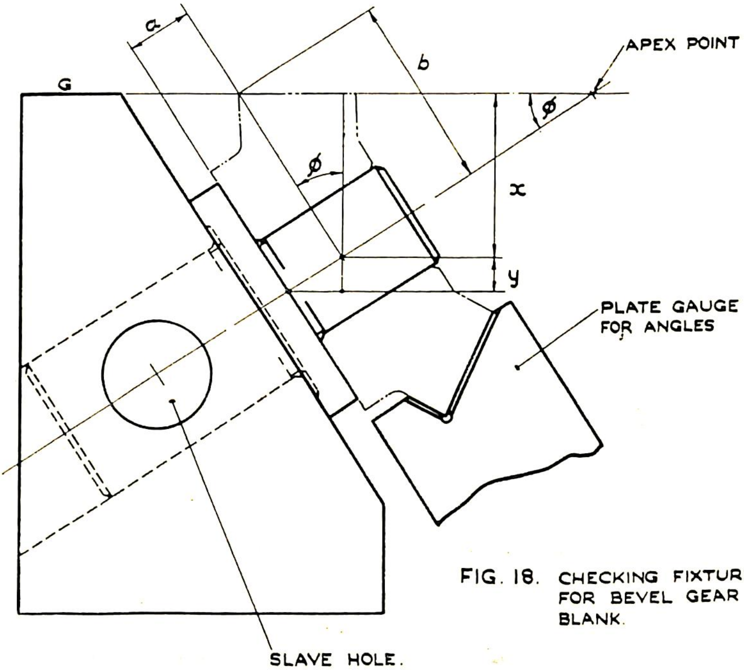

FIG. 18. CHECKING FIXTURE FOR BEVEL GEAR BLANK.

If the gears have to be hardened and ground, then a further dummy should be provided, the dimensions of which should conform to the gear blank, and the tip distance dimensions given on the finished drawing.

These blanks are then used to check any testing equipment provided to check the gears during the various machining operations.

Another reason for making the blanks to very close limits is that gauges can be designed to locate off some part of the blank to check other dimensions such as the tip distance, but if the blank sizes were not reliable, then it would be an impossibility to establish the tip distance in relation to the gear blank.

A simple type of fixture used to check the gear blanks is shown in Fig. 18, and to obtain the gauging dimension $(x + y)$ use is made of the outside diameter and the tip distance dimensions.

A dial indicator is set at zero on the top face of the gauge marked G, and the gear is correct if the same reading is obtained on the face angle of the gear.

To obtain dimensions x and y we have

$x = b \cos \phi$ and $y = a \sin \phi$.
when b = outside radius of gear blank to tip.
a = distance from back face to tip.
ϕ = face angle of gear blank.

The dimensions x and y are then added together, but to manufacture the gauge it will be necessary to provide an accurately produced slave hole as shown on figure.

To avoid making a fixture for every gear an adjustable fixture could be made similar to that shown in Fig. 19, but for mass production the small fixtures are preferred.

The adjustable fixture however will be found very useful for small lots, and ensure that the blanks are being produced accurately prior to the gear cutting operation.

The back face angle should also be checked from the front face angle, and for this purpose a small plate gauge is used as shown in Fig. 18.

By using the method of gauging described, an accurately machined blank will be obtained, or one having the correct cone distance on which the teeth can be cut to drawing sizes. Thus when assembling the gears it will be found that the back angles of the gears will be flush when set at the correct running position.

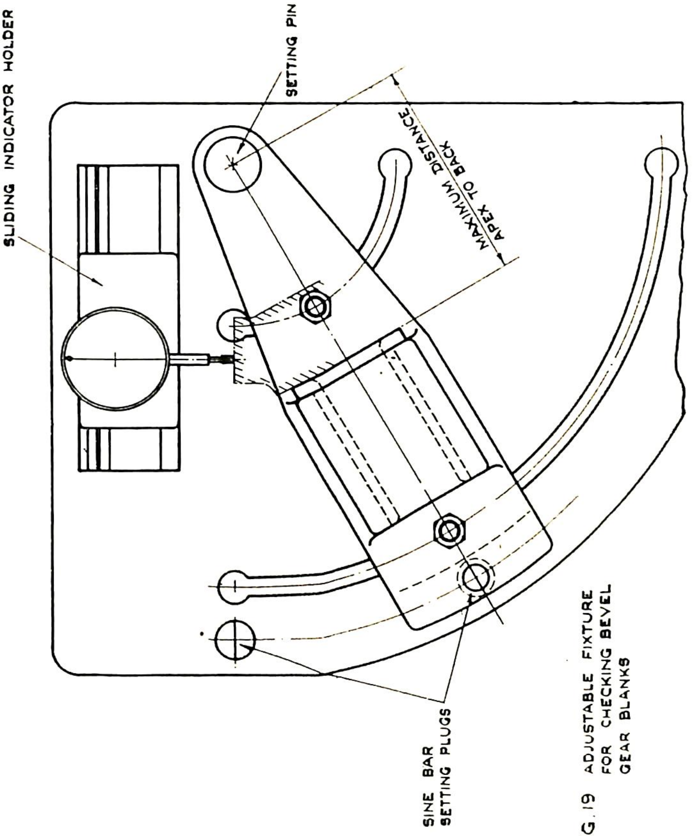

FIG. 19 ADJUSTABLE FIXTURE FOR CHECKING BEVEL GEAR BLANKS

Checking Teeth.

To check the tooth sizes on the back angle face, gear tooth verniers are used, the verniers being set to the chordal addendum and thickness dimensions given on the gear drawing. .

Care has to be taken, however, to see that the check is taken on the back angle face, and it is advisable to slide the verniers from front to back when checking, *i.e.*, with the verniers set on low limit they should not be able to slide over the back end of the tooth, but if set on high limit they should slide over if the tooth thickness is within limits.

This method is however not really satisfactory, and for mass production, master gears are used to check the tooth sizes in the inspection department.

These master gears are made with a nominal chordal thickness dimension, or in other words without any backlash allowance, therefore if a mating pair of master gears were set at their correct relative apex distances they should rotate smoothly, and without backlash. To make this check a centre testing machine similar to the one used for spur gears, but with the addition of a special bracket, is used, generally as shown in Fig. 20, and if the tooth form is incorrect the error will also be shown on the clock indicator by the amount of kick-off.

A common error often made with the use of these machines is to run two production gears together at close centres, and kick-off readings taken, when in actual fact the apex points of both gears do not meet on a common apex in this position.

False readings are therefore obtained, and the only way to check production gears on this type of machine is to turn the gears by hand with the gears in their correct running positions, *i.e.*, with the apex point of both gears meeting on a common apex, and detect the kick-off by feel.

A more reliable method however is to use a Gleason testing machine. Mount the gear to be tested in the driven spindle, and run it with a master gear mounted in the driving spindle under light load.

In this way the running conditions can be observed on the flanks of the gear teeth, such as the location of the bedding, and if found to be unsatisfactory the necessary adjustments can then be made on the gear cutting machine.

This test should be carried out after each sharpening and re-setting of the cutters, but a concentricity check should be taken on every gear as it comes off the gear cutting machine, and also

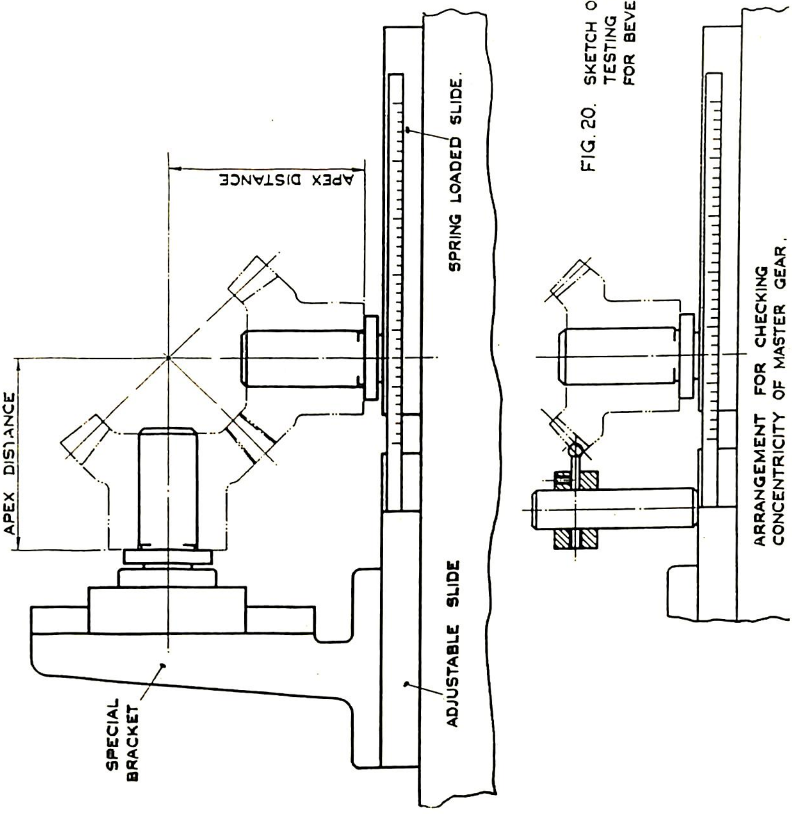

FIG. 20. SKETCH OF CENTRE TESTING MACHINE FOR BEVEL GEARS.

ARRANGEMENT FOR CHECKING CONCENTRICITY OF MASTER GEAR.

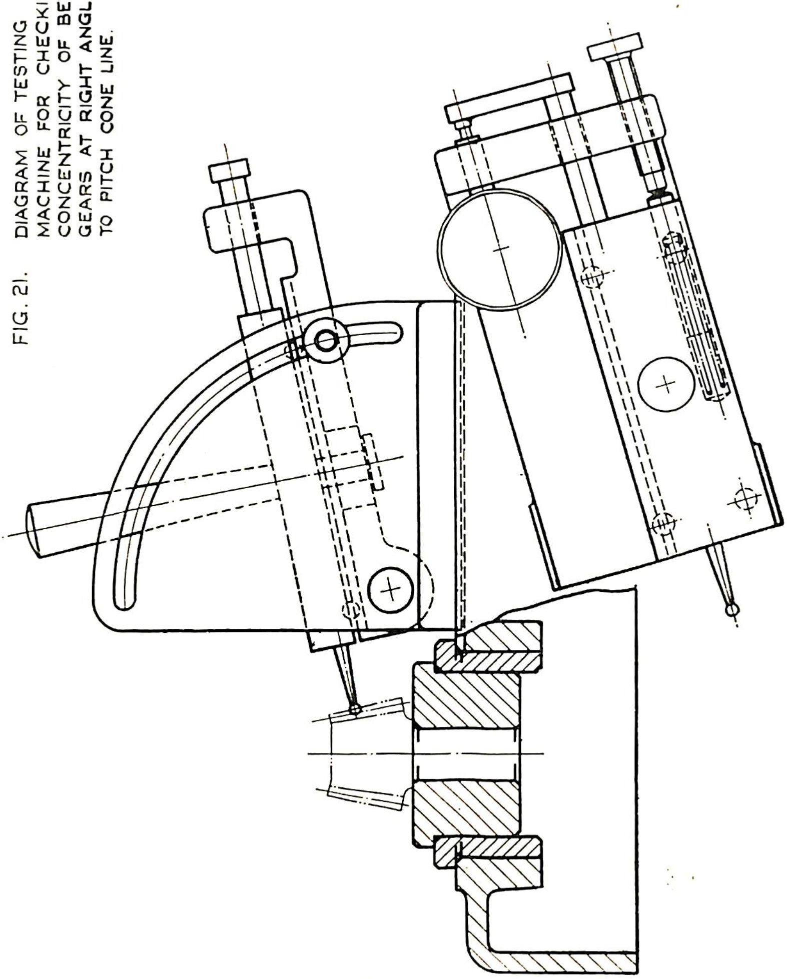

FIG. 21. DIAGRAM OF TESTING MACHINE FOR CHECKING CONCENTRICITY OF BEVEL GEARS AT RIGHT ANGLES TO PITCH CONE LINE.

after finally grinding the bore in the case of a hardened gear. For this check a ball point is mounted on the adjustable slide of the testing machine, the gear to be tested being mounted on the other slide, as shown in Fig. 20, and readings obtained from the dial indicator.

In some cases, however, special testing machines are designed for the concentricity check, and in this case the ball point is applied at right angles to the pitch cone line, and not at right angles to the axis of the gear, which is the case when the centre testing machine is used.

A diagram of a testing machine used for checking concentricity with the ball point applied at right angles to the pitch cone line is shown in Fig. 21.

To obtain accurate master gears the teeth should be cut after final grinding of the shank or bore, and for this purpose the gears are made of Meehanite in some cases. Some firms, however, prefer hard gears, and nitralloy steel is the most suitable as there is less risk of distortion during the hardening operation than would be the case if ordinary case hardening steel was used.

A very simple type of fixture used to check bevel gears in conjunction with a master gear is shown in Fig. 22.

This fixture ensures that the gear will run correctly at its proper apex distance, but does not definitely check eccentricity, although if this does exist it can be detected by the varying amount of backlash, or a possible tightness in the meshing at the high point.

This type of fixture replaces the centre testing machine shown in Fig. 20, and as one is required for each gear it also obviates any possible chance of error when setting the universal machine.

Before sending the gears to stores it is advisable to etch or stamp on the gear the apex distance, which can be obtained from the fixtures, Figs. 20 and 22, in order that the gears can be mounted at the same distance on assembly.

Facilities should be provided, however, to measure this distance on the assembly before the gears are fitted, any difference being made up by a series of shims.

The design of the gear casings should also be such that the actual measuring is accessible and that the back angle faces of the gears can be seen when correctly mounted to make sure that they are flush. For this purpose a sighting hole is sometimes provided. Afterwards it is plugged up to seal the housing against oil leakage.

Bore Concentricity.

Hardened gears are much more difficult to produce than soft gears, as location diameters which were concentric to the gear teeth before hardening, have to be re-machined after heat treatment.

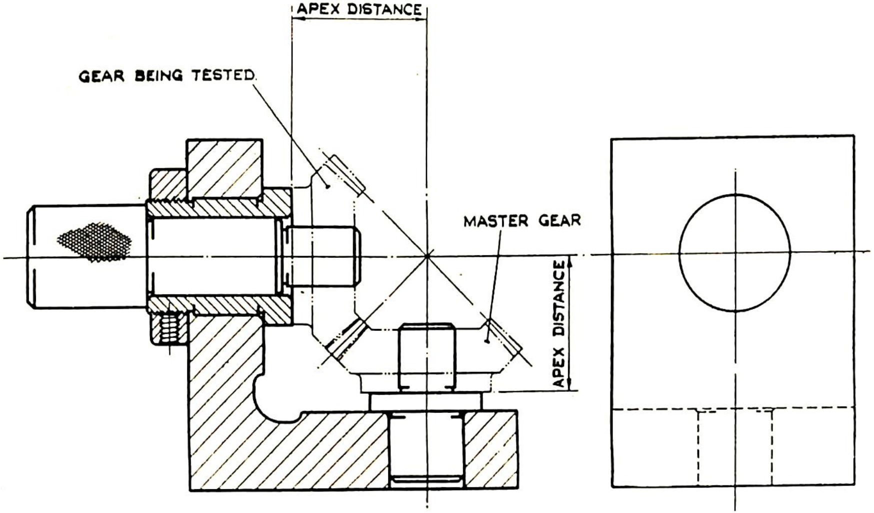

FIG. 22. SIMPLE CENTRE TESTING FIXTURE FOR BEVEL GEAR MESHING WITH MASTER GEAR.

Bevel gears made integral with the shaft, in which centres are provided, do not present the same difficulties as bored gears, due to the fact that the same centres can be used for final grinding as were used for grinding the locations used for the tooth cutting operations.

For bored gears therefore it is advisable to locate from the teeth of the gear in a special fixture, in order to grind the bore concentric with the teeth. Various types of these fixtures have been tried, some locating from the face angle of the gear blanks, others in which the teeth are located on fixed taper pins, but in all cases the results obtained are not entirely satisfactory.

The whole difficulty lies in the fact that the fixtures described are very difficult to produce within the limits of the accuracy required, therefore a fixture should be provided in which the accuracy of the fixture is not dependant on producing an accurate component. A fixture having these features and which has been used with very satisfactory results is shown in Fig. 23.

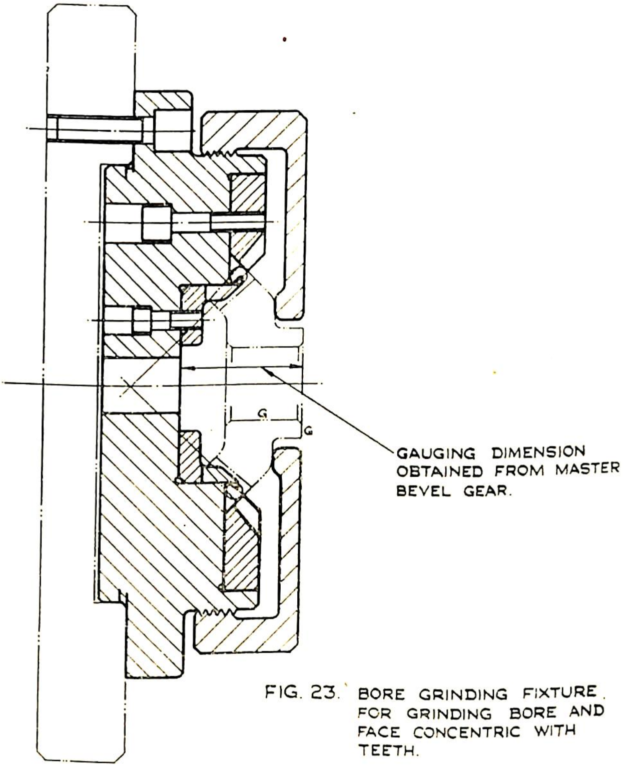

FIG. 23. BORE GRINDING FIXTURE FOR GRINDING BORE AND FACE CONCENTRIC WITH TEETH.

The gear teeth contact the balls, one of which is roughly located in each tooth space or alternate tooth space by means of the cage, which also prevents the balls from falling out.

By locating in this manner the balls are pushed against the flat, and inclined hardened and ground faces, which can be easily and accurately produced, thus locating the teeth concentric about the centre of the machine.

The gear should be lightly clamped by a ring nut as shown in figure or by a central pull rod from the back arranged to pull three floating clamps.

The clamping is really important, as it would be inadvisable to use three independent clamps, each having a tendency to tilt the gear before the other is tightened.

The back location face of the gear should also be ground at the same setting, and the master gear first placed in the fixture, in order to determine a gauging dimension from a fixed point to the back face, a clock indicator depth gauge being found very useful for the purpose.

Quenching Dies.

To keep bevel wheels of both straight and spiral type flat during the quenching operation a special quenching machine is used. Special dies are made for each gear, and these hold the gear flat and concentric while being quenched, within a limit of 0·002.

Providing this is done the gear can then be held lightly in a soft jaw chuck and the back face and bore checked with a clock indicator before grinding to an allowable tolerance of 0·001 clock reading.

The finished gear will then be found to be within the limits of 0·002 when checked for concentricity using the ball point method previously described and shown in Fig. 21.

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3. Deflection of Shafts and Beams (Instruction Sheet).
4. Steam Radiation Heating Chart.
5. Horse-Power of Leather Belts, etc.
6. Automobile Brakes (Axle Brakes). } Connected.
7. Automobile Brakes (Transmission Brakes).
8. Capacities of Bucket Elevators.
9. Valley Angle Chart for Hoppers and Chutes.
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